POWER

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A man on the job—ready for any emergency—fully aquainted with every piece of machinery in his charge—fully alive to the possibility of his department.

A man who runs his plant intelligently, so that every dollar spent in his department yields a dollar's worth of value and maybe more.

A man who knows that his department is operated on as economical a basis as is possible with modern equipment.

A man who is not worried by central station figures—who knows he can hold his own.

A man who can prove to his boss at any time that he is the right man for his job.

A man who goes about his work with a smile, who handles his men with a smile, who is master of himself and his future, who takes advantage of every opportunity to add to his knowledge of power engineering—who knows things, in other words, instead of guessing—who does things instead of wondering how to do them.

That man is a live wire.

Do you do your work with a confident smile? Are you a live wire? Be one!

Savannah Electric Co.'s New Power Plant

By Warren O. Rogers

SYNOPSIS—This new, 11,000-kw. capacity, steam-turbine central station is probably one of the most uptodate installations in the South. It will displace the old reciprocating-engine, belt-driven units of the old plant. The boilers are on a floor above the turbines, and are separated from them by a brick wall. Controlling switches, electrical recording instruments, etc., are on galleries on the opposite side of the turbine room.

In order to economically supply the demand for electrical energy in the city of Savannah, Ga., which has outgrown the capacity of the old plant, a new, modern installation has been built and put into operation.

The old plant was, at one time, a good example of the modern central station of its period, and that not so many years ago. The generators were belt driven from

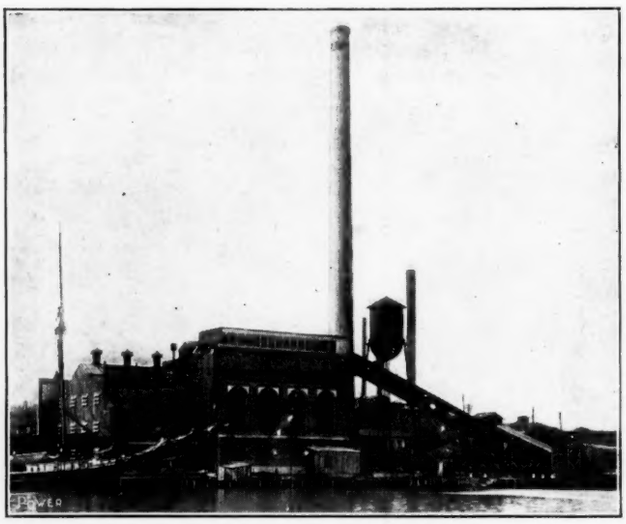


Fig. 1. View of the Savannah Electric Co.'s New Power Plant from the River

reciprocating engines, and both return-tubular and water-tube boilers were used. The switchboard and controlling apparatus, then modern, are now out of date. Most of the electrical apparatus has been removed and all electrical energy generated by the company will soon be sent out from the new plant.

Savannah, like several other Southern cities, has two electric-lighting companies, and as a result, competition has made the price to users of electrical energy considerably below that of cities where but one company supplies the demand. It is understood that the price ranges from 6c. per kilowatt-hour down.

The new plant is located on the banks of the Savannah River, Fig. 1. It is a brick building and the general interior design attracts the eye at first glance. Entering the building from the street, one steps into a hall, off of which the chief engineer's office is located. A stairway leads from the inner end of the hall to the lower or switchboard gallery, on the right-hand side of the plant. Above this gallery is a second, used for purposes to be described later. Both of these galleries open on the turbine room. On the opposite side of the plant is a bay on a level with the turbine floor, which extends under the boiler setting. Beyond the dividing wall of the bay is an ashroom, above which is the boiler-room firing alley. Between the ashroom and the bay in the turbine room is the brick setting of the boiler ashpits. One end of the boiler and turbine room is made with temporary walls to allow further extension of the plant when the demand justifies such action.

Upon entering the turbine room, the visitor obtains a general view of the generating units, consisting of two 5000-kw. and one 1000-kw. turbine units. The turbines operate with 180 lb. steam pressure superheated 125 deg. Both large units run condensing at 1800 r.p.m. The generators of these turbines are of 6250 kv.-a. capacity, and produce two-phase, 60-cycle current at 2300 volts. The 1000-kw. turbine drives a 1250-kv.-a. generator.

Contrary to common practice, the condensers, instead of being in a basement or condenser pit are on the floor level with the turbines. All auxiliary units are also on the same floor level. This arrangement brings all of the units of each main generator where their operation is under the immediate observation of the man on watch, and also renders the crane available so that repairs or inspections can be made most expeditiously.

The turbines are placed crosswise of the room and the surface condenser and circulating-pump units are beside them. A general view of the turbine room is shown in Fig. 2. The arrangement of the surface condensers and other auxiliary machinery is also shown. Fig. 3 shows a plan and elevation of the entire plant.

Each large surface condenser contains 13,000 sq.ft. of cooling area. The small condenser has 4000 sq.ft. of cooling surface; all condensers are of the horizontal type.

Circulating water is obtained from the Savannah River and is supplied to the large condensers by two 18-in. volute pumps, directly connected to 11x12-in. vertical steam engines. The small condenser is supplied with circulating water by a 10-in. volute pump directly connected to a 6x7-in. vertical steam engine.

On the turbine floor at the end of each of the larger water-circulating units is a vertical steam turbine directly connected to a centrifugal hotwell pump; running at 2000 r.p.m.

The dry-vacuum pumps operating in connection with the large condensers are placed at the head of each unit. They are 10 and 26 by 18 in. in size. The small vacuum pump is 8 and 16 by 12 in. in size.

Exciter Units and Gage Boards

Two exciter units are driven by 125-kw. horizontal steam turbines at 2400 r.p.m. and run noncondensing. Each turbine is directly connected to a direct-current generator of 1000 amperes capacity, at 125 volts. The vacuum pumps and exciter units are on the south side of the plant.

At the head of each turbine is a gage board upon which are mounted a frequency meter, an indicating steam gage, an indicating spring-vacuum gage, an absolute mercury-pressure gage, a recording steam-temperature gage to show the temperature of the inlet steam and a record-

ing vacuum gage connected to the exhaust to the condenser. The gage boards for the two larger turbines also contain a spring gage which indicates the pressure in the first stage of the turbine.

Pumps

Fig. 2 shows the location of the pump bay; a near view of the pump units is shown in Fig. 4. They consist of a 16 and 10 by 18-in. pot valve, outside-center packed, duplex, boiler-feed pump, one 7½ and 4½ by 6-in., duplex, outside-end packed, pot-valve pump, used for boiler feed, and one turbine-driven unit, the water end of which is a 4-in., four-stage, centrifugal pump, also

motor-generator set for supplying railway current. The synchronous motor is of 1450 hp. capacity, using 2300-volt, 60-cycle current, and is built for 80 per cent. power factor leading current. The generator end delivers 600-volt current and is of 1667 amperes capacity, at a speed of 514 r.p.m.

A study of the turbine room shows that ample space has been provided about each unit which will greatly facilitate the work necessary about the plant when repairs are made. The floor is of concrete. The roof is also of concrete with tar and gravel covering supported by steel roof trusses carried on the main binding. A 45-ton crane serves the entire area of the turbine room.

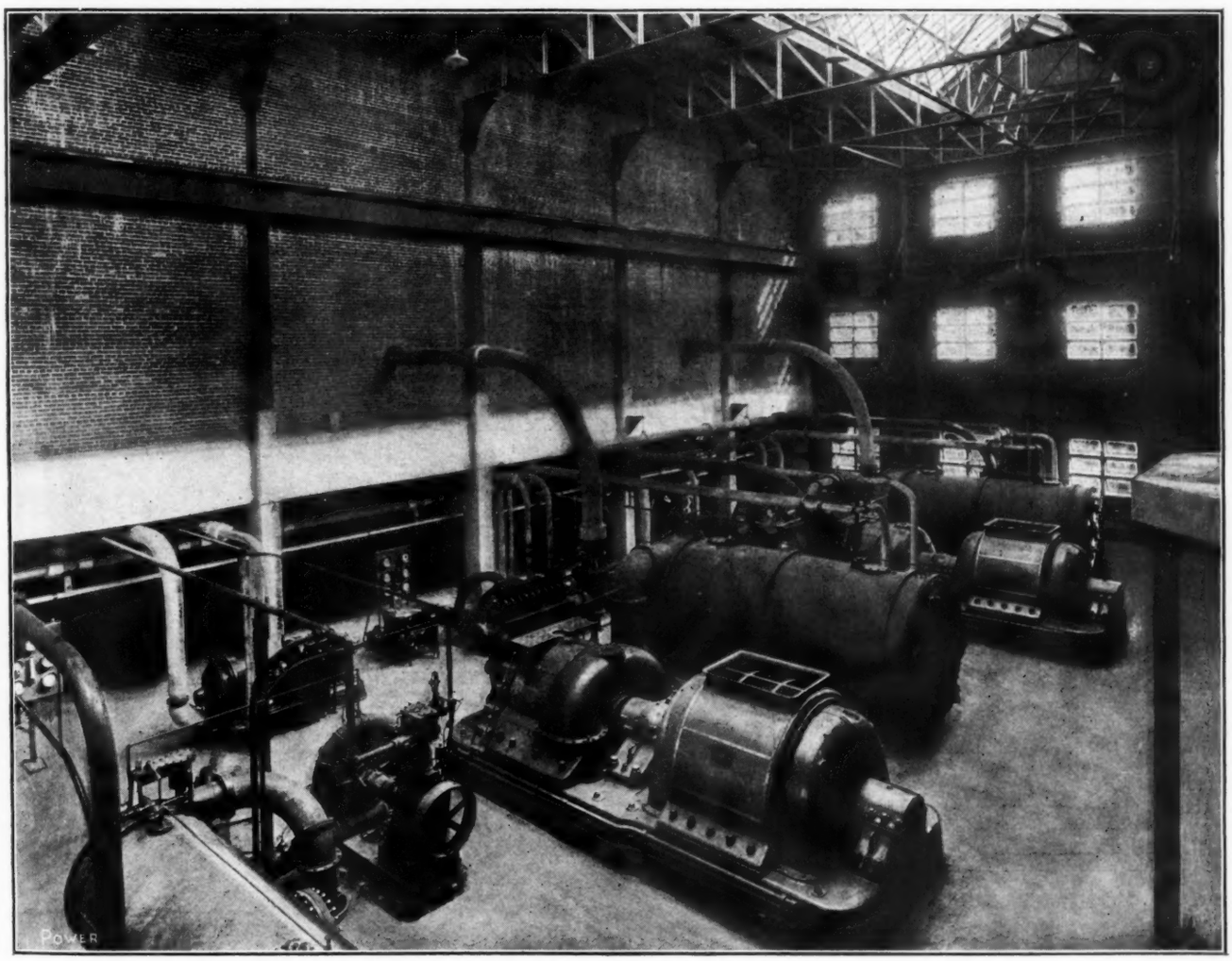


Fig. 2. General View of the Turbine Room

used for boiler feed. This gives a flexible means of supplying boiler-feed water. A 7½ and 8½ by 10-in., piston-pattern, duplex pump is used for low-service duty. A 3-in., 175-r.p.m., single-stage centrifugal pump is directly coupled to a 15-hp., two-phase, 220-volt induction motor. This is used for pumping water to the elevated tank and for other station uses. This pump is normally operated continuously but is relayed by the 7½ and 8½ by 10-in. steam-driven pump already mentioned.

A small 1½-hp., motor-driven, 1½-in. centrifugal pump handles the raw water supplied to the settling and filtering tanks.

At the end of the room next to the small turbine, is a

Boiler Room

The arrangement of the boilers is of interest, in that the greater part of the ground area beneath them is used to advantage for purposes other than that of ashpits. The floor space devoted to pumps is in the bay under the rear end of the boilers and gives just that much more floor area to the turbine room. All headers and pipe connections are either placed back of the boilers in the boiler room or are suspended from the ceiling of the pump bay.

There are four 708-hp., three-drum, water-tube boilers, each equipped with superheaters which superheat the steam 125 deg. F. The boilers each have 294 four-inch

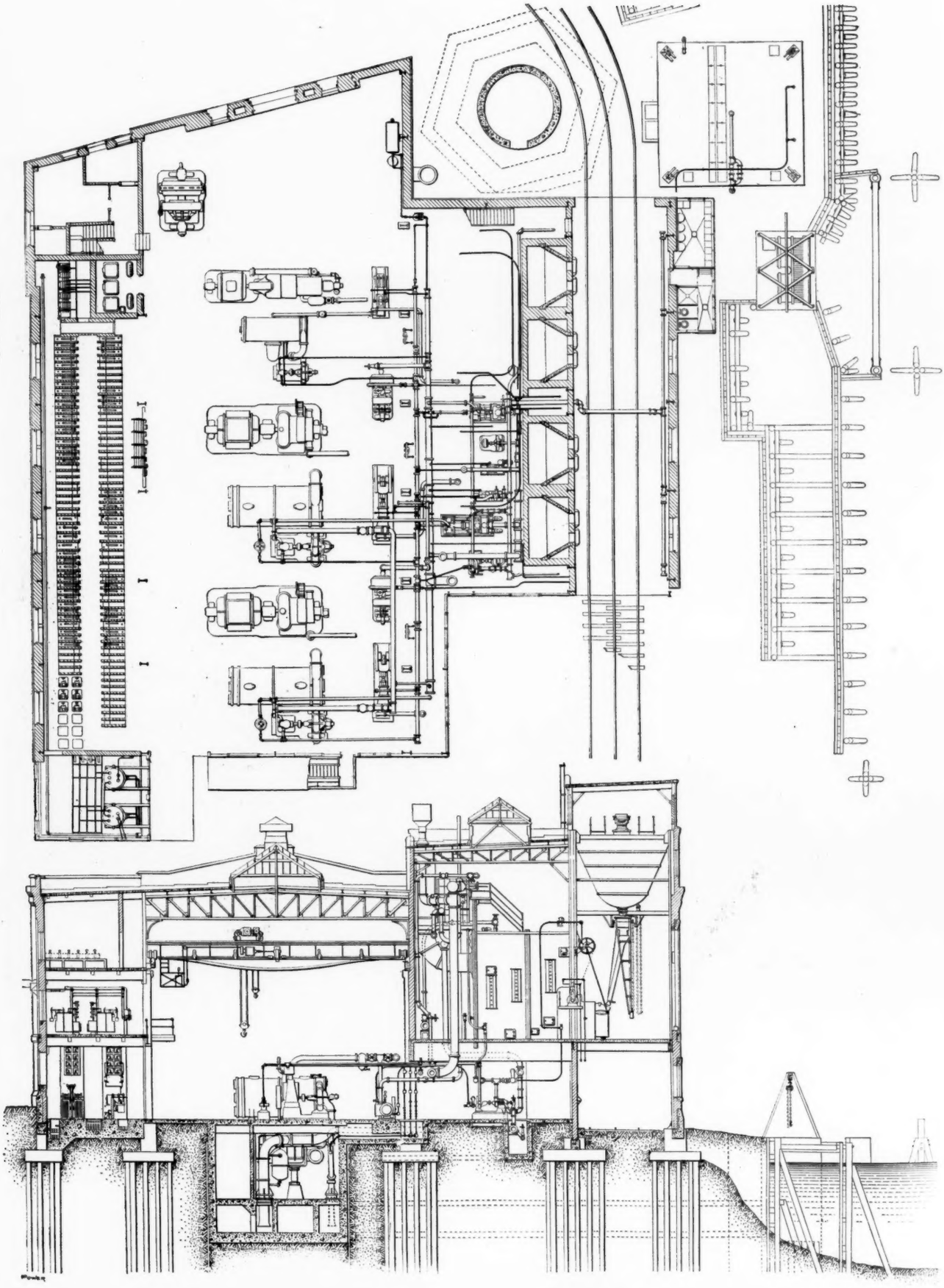


FIG. 3. PLAN AND ELEVATION OF THE SAVANNAH POWER PLANT

tubes, 20 ft. long, and three 42-in. drums made of ½-in. plate; they are 22 ft. 3 in. long, and are made of open-hearth steel having a tensile strength of 56,000 lb. The boilers are designed for 200 lb. steam pressure. Steam is measured by steam-flow meters, one for each boiler.

Chain-grate stokers, each 11x12 ft., or 132 sq.ft. of active grate area, are used. They are driven by eccentric and rods from an overhead shaft hung from brackets on

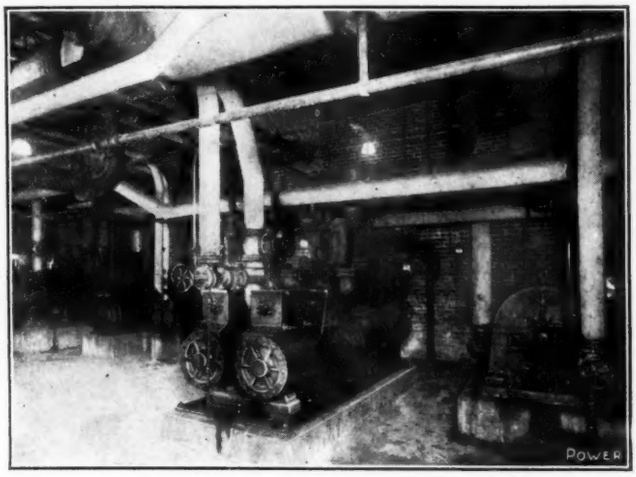


FIG. 4. PARTIAL VIEW OF THE PUMP ROOM

the front of the boilers. The shaft is belt driven by either of two vertical steam engines, one placed between the two batteries of boilers, the other at the end of the second battery next to the temporary end partition. The shaft is provided with a clutch at each belt wheel, so that a section of the shafting can be cut out if desirable, or each battery of boilers can be operated independent of the other stoker engine. A partial view of the boiler room from the front is shown in Fig. 5.

Coal is delivered to the stokers from a 320-ton capacity, steel, concrete-lined, suspended coal bin, parabolic in shape. It extends the length of the boiler room and holds several days' supply of fuel. The concrete lining protects the steel casing from any corrosive action of the coal. A coal chute, equipped with a weigher, travels on a track suspended from the under side of the storage bin. The weigher is filled from either one of the four outlets of the storage bin, and the fuel is delivered to the stoker hoppers after being weighed.

Fuel is usually brought to the plant in vessels, and either unloaded on a pile near the wharf by a tub hoist, or is hoisted and dumped into a hopper above the crushing rolls, from which it falls onto a belt coal conveyor, running on an incline from the coal crusher on the wharf to a point central with the storage bin in the boiler house. Here it discharges the coal onto a second belt conveyor running at right angles to the first, by which the fuel is delivered to the coal bunker. In case no cargo is to be unloaded, the necessary fuel to keep the plant in operation is hoisted from the storage pile in the yard and dumped into the crusher, and conveyed to the coal-storage bin over the boilers. Rail coal can also be handled.

A view over the top of the boilers is shown in Fig. 6. The tops of the boilers have been filled in level, so that there is a smooth, flat surface, with the exception of the pipe connections and the pockets made for convenience in getting at the top row of tubes through doors at the bottom of the clean-out spaces, two of which are shown in the foreground of Fig. 6. All piping above the boilers is shown, and its simplicity contrasts with the array found in many boiler installations. There are three pop safety valves for each boiler, two on the saturated-steam pipes and one on the superheated-steam connection.

Feed water can be drawn from the open heater, seen at the right-hand corner of Fig. 6, and pumped to the boilers through either a 6-in. main feed line feeding in at the front of the drum, or through an auxiliary 1½-in. feed line at the rear. Due to the elevated position of the heater, the water goes to the feed pumps by gravity. In case of emergency the water can be cut out and water drawn directly from the river. A covered steel smoke flue runs along the rear ends and above the setting of the boilers. A smoke uptake from each boiler connects with the bottom of the main smoke flue, as shown in Fig. 7. The draft is controlled by a damper regulator, the damper being placed in the smoke flue between the boiler and the reinforced-concrete chimney, which is 10 ft. inside diameter at the top and is 253 ft. 6 in. high.

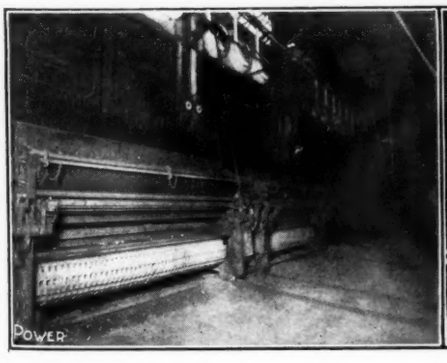


FIG. 5. FRONT OF BOILERS

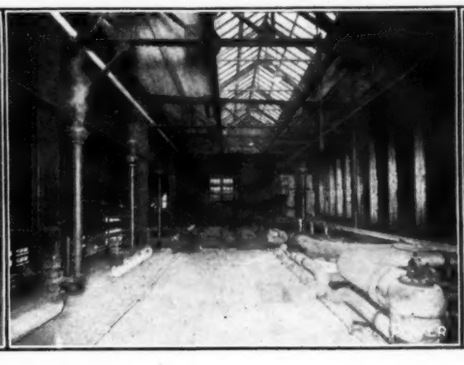


FIG. 6. SETTING OVER BOILERS

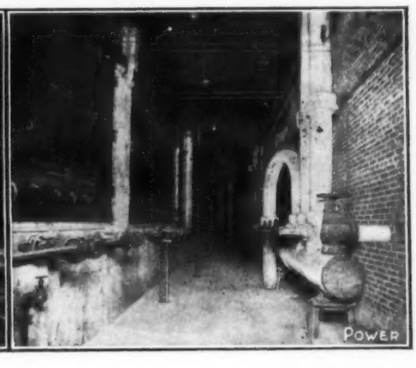


FIG. 7. BACK OF BOILERS

The 3-in. blowoff header, Fig. 7, has three connections to each boiler, with a blowoff valve between the boiler and the fitting in the header. At the end of the header an angle valve connects with a vertical blowoff pipe, as shown.

PIPING

Steam is conveyed from the boiler to the 12-in. main steam header at the rear of the boilers through four 6-in. long-radius bent pipes which run in a horizontal di-

rection above the boilers and then drop vertically to the header, as shown in Figs. 6 and 7. This header rests on roller saddles, about 3 ft. above the boiler-room floor. A U-expansion bend is connected to the header at the far end, just beyond the last boiler. This position will bring

FIG. 8. SWITCHBOARD GALLERY AND BUSBAR COMPARTMENTS

it in the center of the header when duplicate boiler units have been installed.

The steam pipes leading from the boilers to the header are fitted with two stop valves, one automatic nonreturn valve close to the boiler outlet, the other just above the header. A gate valve in the header permits of cutting out one-half of the header, two boilers supplying steam to each half, if necessary. The main 12-in. header is connected to the auxiliary 6-in. header, hung from the ceiling of the pump bay, by a 6-in. pipe connecting at each end of the header. Both are piped from the top of the main header and, making a U-bend, drop through the floor to the auxiliary header. All valves and fittings in the superheated-steam piping system are of openhearth steel.

All main turbine units are piped directly to the main 12-in. header. The small 1000-kw. unit is supplied with steam through a 6-in. long-radius bent pipe; the two large turbines through an 8-in. long-radius pipe, as shown in Fig. 2. These connections are made from the top of the header through an angle valve, thus avoiding pockets for the accumulation of water when the units are shut down, Fig. 7.

Steam for the auxiliaries is taken from the top of the auxiliary 6-in. header, and extended in either direction horizontally to the various pumps, etc., dropping with short-radius bends, as shown in Figs. 2 and 4.

An exhaust-steam header is also hung from the ceiling of the pump-room bay. It begins with 8 in. at the end next to a small air compressor at an end of the bay and is enlarged to 14 in. at the other end. A 16-in. pipe beyond the vertical 20-in. connection to the heater permits of handling exhaust steam from future auxiliary units. The 20-in. exhaust pipe runs to the heater, which can be bypassed, the exhaust steam escaping to the atmosphere through a 16-in. horizontal back-pressure valve. The live-steam piping is drained through a drip header to the Holly receiver and discharge chamber and the water returned to the boilers by gravity.

Makeup feed water for boiler use is taken from the Savannah River. On account of the muddy condition of the water, it is first admitted to a fluctuating tank placed below the ground level, into which a coagulating solution is pumped. When the mud has been precipitated, the water goes through a filter into a storage tank from which it is pumped to the feed-water heater.

Notwithstanding that the arrangement of the various units has been such that ample floor room has been provided around them, the same liberal space has been devoted to the electrical-controlling devices, etc.

PRINCIPAL EQUIPMENT OF THE SAVANNAH ELECTRIC CO.'S NEW POWER PLANT

No.	Equipment	Type	Purpose	Kv.a.	Kw.	Phase	Cycles	Volts	Amp.	Hp.	R.p.m.	Size	Steam Press. Lb.	Manufacturers
1	Turbine	Horizontal	Main units	....	1000	....	....	....	....	....	1800	.....	180	Allis-Chalmers Co.
2	Turbines	Horizontal		....	5000	....	....	....	....	....	1800	.....	180	General Electric Co.
1	Generator	Alt. current		1250	....	2	60	2300	....	....	1800	.....	...	Allis-Chalmers Co.
2	Generators	Alt. current		6250	....	2	60	2300	....	....	1800	.....	...	General Electric Co.
1	Condenser	Surface		....	....	....	....	....	....	....	....	4000 sq.ft.	...	Henry R. Worthington
2	Condensers			....	....	....	....	....	....	....	....	13,000 sq.ft.	...	
1	Pump	Volute	Circ. water	....	....	....	....	....	....	....	300	10"	...	B. F. Sturtevant Co.
2	Pumps			....	....	....	....	....	....	....	225	18"	...	
1	Engine	Vertical	Circ. pumps	....	....	....	....	....	....	....	300	6x7"	180	
2	Engines			....	....	....	....	....	....	....	225	11x12"	180	
2	Turbines	Vertical	Hot-well pumps	....	....	....	....	....	....	....	2000	.....	180	Terry Steam Turbine Co.
1	Pump	Horizontal	Dry air	....	....	....	....	....	....	....	100	8x16x12"	180	Henry R. Worthington
2	Pumps			....	....	....	....	....	....	....	100	10x26x18"	180	
2	Turbines	Horizontal	Exciter units	....	125	....	....	....	....	....	2400	.....	180	General Electric Co.
2	Generators	Direct current		....	125	....	....	125	1000	....	2400	.....	...	
1	Pump	Pot valve	Boiler feed	....	....	....	....	....	....	....	....	16x10x18"	180	Henry R. Worthington
1	Pump	End packed		....	....	....	....	....	....	....	....	7½x4½x6"	180	
1	Turbine	Curtis		....	....	....	....	....	....	....	....	.....	180	General Electric Co.
1	Pump	Four-stage		....	....	....	....	....	....	....	....	4"	...	Alberger Pump & Condenser Co.
1	Pump	Duplex	Low service	....	....	....	....	....	....	....	....	.....	180	Henry R. Worthington
1	Pump	Centrifugal		....	....	....	....	....	....	....	175	3"	...	
1	Motor	Induction	Centrifugal pump	....	....	2	60	220	....	15	1800	.....	...	General Electric Co.
1	Motor-generator	Synchronous	Motor service	....	....	2	60	2300 / 600	1667	1450	514	.....	...	
1	Crane	Case	Turbine room	....	....	....	....	....	....	....	....	45 tons	...	Case Crane Co.
4	Boilers	Water tube	Steam gen.	....	....	....	....	....	....	708	....	.....	180	Babcock & Wilcox Co.
4	Superheaters	Foster	Steam	....	....	....	....	....	....	....	....	.....	...	Power Specialty Co.
4	Meters	Steam flow	Steam	....	....	....	....	....	....	....	....	.....	...	General Electric Co.
4	Stokers	Chain grates	Furnace	....	....	....	....	....	....	....	....	132 sq.ft.	...	Green Engineering Co.
1	Conveyor	Belt	Coal	....	....	....	....	....	....	....	....	.....	...	Robins Conveying Belt Co.
12	Valves	Safety	Pop	....	....	....	....	....	....	....	....	.....	...	Ashton Valve Co.
1	Heater	Open	Feedwater	....	....	....	....	....	....	....	....	.....	...	Exeter Machine Wks.
	Switchboards	Panel	Elec. control	....	....	....	....	....	....	....	....	.....	...	General Electric Co.

ELECTRICAL APPARATUS

The main distributing switchboard and the benchboard for control of generating and converting equipment are on the first gallery opposite the boiler room, as shown in Fig. 8. A double 2300-volt bus in concrete and brick

compartments is located over the oil switches in the bay below the switchboard gallery. The bus structure is in two rows, at the end of which are located the feeder regulators.

The five-panel switchboard shown at the right of Fig. 8 on the generator floor controls the station lighting and power service. Fifteen mercury arc sections for street lights are located in the switchboard gallery.

On the top balcony the turbine-field rheostats are placed, with provision for four more. This floor also affords room for a storage battery for operating the solenoid control switches in the station. A projection, central with the length of the switchboard gallery, affords means for the operator to obtain a view of the entire turbine room. The balconies and floor of the boiler and turbine room are of concrete. The building is well illuminated, both by natural light during the day and artificial illumination by night.

Distributing circuits are carried from the station overland with the exception of three line cables for operating two substations. These are taken out underground.

When the new substation, now nearing completion, is put into service, the electrical energy from the power plant will be transmitted to it in a duplicate underground line, there to be converted by motor generators for supplying an Edison three-wire system in the business district. This new building will contain the necessary converting apparatus and offices of the company. It is located in a district convenient for business and distributing purposes. This power plant was designed and constructed by the Stone & Webster Engineering Corporation, Boston, Mass.

The "Flat-Curve" Engine

By A. D. Skinner

SYNOPSIS—Valve leakage rather than increased cylinder condensation is the cause of greater steam consumption at light loads.

For years the astrologer's stone in steam engineering has been the "flat-curve" engine, i.e., an engine that would give the same steam consumption per indicated horsepower-hour over a wide range of load. The profession has become so used to curves that begin to mount up or show a "heel" at half load that they have begun to accept it as the result of a condition impossible of improvement, and have attributed the increased steam consumption at the light loads to cylinder condensation. An attempt will be made to prove that the cause of the much greater steam consumption at light loads is not increased cylinder condensation, but merely valve leakage.

Fig. 1 shows the economy obtained from a 13x15-in. Skinner engine, operating noncondensing, with a steam pressure of 104 lb. gage at the throttle, the speed of the engine being 268 r.p.m. Saturated steam was delivered at the throttle, having an average moisture of 2.2 per cent. by throttling calorimeter, which was allowed for

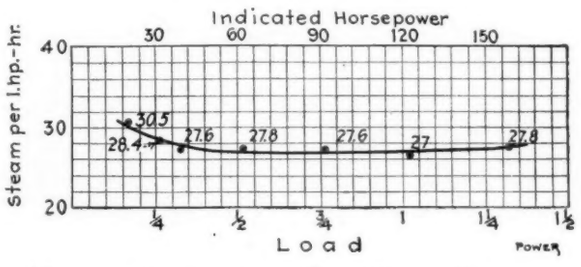


Fig. 1. Results from Flat-Curve Engine

in computing the net steam consumption. The tests were made by condensing the exhaust at atmospheric pressure in a surface condenser, and weighing the condensate. Several tests were made at the light loads to check the correctness of the former tests and to establish definitely the curve.

The valve was of the Skinner standard steam-tight type, and was proportioned to obtain the desired results. It was not reset or adjusted as to lap or lead during the entire series of tests.

The curve is remarkable for its flatness, and is unique in steam-engineering practice for any engine, whether single-valve or multiple-valve, as it shows practically the same steam consumption per indicated horsepower-hour from ¼ to 1¼ load inclusive. The usual textbook

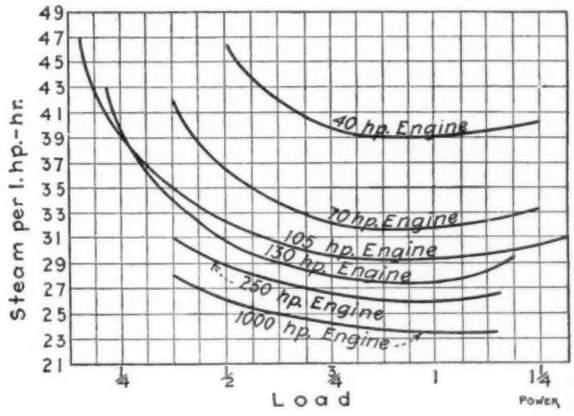


Fig. 2. Typical Rate Curves of High-Speed Engines

theories of cylinder condensation for light loads seem to be quite set at nought by the results of these tests.

Fig. 2 is reproduced from a widely read textbook on steam engines, for the reason that these curves are designated in this textbook as "typical economy curves of high-speed engines." These curves all show the usual heel at loads less than half.

Increased cylinder condensation, due to early cutoffs, assumes relatively little importance as compared with the steam-tight valve; and practically all tests to determine the amount of cylinder condensation have been based upon the performance of engines having leaky valves, whether these engines were single-valve or multiple-valve.

The builders of pressure-plate valve engines have until recently stated in their catalogs that there was 0.003-in. clearance between the valve and the housings. Therefore, there is an initial leakage, which, of course, increases

the longer the engine is operated. To prove these statements, take the guarantees made recently by a prominent engine builder on a pressure-plate valve engine. This engine was of twice the capacity of the engine that produced the flat curve. The steam pressure was to be 90 lb., whereas the flat-curve engine had a steam pressure of 104 lb.; but the pressure-plate valve engine, being of twice the capacity, had an advantage that is not offset by the slightly lower steam pressure. Therefore, the economies of the two engines admit of a fair comparison:

Load	¼	½	¾	1
Pressure-plate valve guaranty	40.9	33.2	30.2	29.8
Flat-curve engine economy	28.4	27.8	27.6	27.0

It may be noticed that the steam consumption guaranty made for the pressure-plate valve engine is exactly 10 per cent. more at full load than the economy obtained by the steam-tight valve engine. Now, the amount of leakage through a valve on an engine may be regarded as constant for all loads. As a matter of fact, it is somewhat greater in amount at light loads, on account of less steam seal, due to early cutoffs, as well as for other reasons which need not be discussed here. But for this case it is favoring the pressure-plate valve engine to assume that it is constant; and the exponents of this type of valve cannot complain when it is assumed for purposes of demonstration, that a pressure-plate valve engine of 120 hp. capacity at 104 lb. pressure (the same horsepower and pressure as for the flat-curve engine) will give just as good economy as was guaranteed for an engine of twice its capacity at 90 lb.

The pressure-plate valve engine guaranty at full load was 29.8 lb., or an hourly steam consumption of 3576 lb. for a 120-hp. engine. The flat-curve engine at full load of 120 hp. gave an economy of 27 lb. per i.hp.-hr., or an hourly consumption of 3240 lb. The difference, 336 lb., or about 10 per cent. of the amount of steam supplied to the steam-tight valve engine, represents the amount of steam leaking past the pressure-plate valve hourly. Remembering that the amount of valve leakage is practically constant for all loads, the processes in Table 1 are worked out to show how this 336 lb. hourly leakage affects the economy of the engine at various loads.

TABLE 1

FULL LOAD

27 lb. × 120 hp. = 3240 lb. per hr., no leakage.
3240 + 336 lb. leakage = 3576 lb. per hr., with leakage
3576 ÷ 120 hp. = 29.8 lb. per hp.-nr.

THREE-QUARTER LOAD

27.65 lb. × 90 hp. = 2488 lb. per hr., no leakage.
2488 + 336 lb. leakage = 2824 lb. per nr., witn leakage.
2824 ÷ 90 hp. = 31.3 lb. per hp.-hr.

ONE-HALF LOAD

27.84 lb. × 60 hp. = 1670 lb. per hr., no leakage.
1670 + 336 lb. leakage = 2006 lb. per hr., with leakage.
2006 ÷ 60 hp. = 33.4 lb. per hp.-nr.

ONE-QUARTER LOAD

28.46 lb. × 30 hp. = 853.8 lb. per hr., no leakage.
853.8 + 336 lb. leakage = 1189.8 lb. per hr., with leakage.
1189.8 ÷ 30 hp. = 39.6 lb. per hp.-hr.

Compare, now, the flat-curve engine economies with 10 per cent. valve leakage added, as above determined, with the guaranteed economies of the pressure-plate valve engine. The rates are almost identical.

Loads	¼	½	¾	1
Pressure-plate valve guaranty	40.9	33.2	30.2	29.8
Flat-curve engine with 10 per cent. leakage.	39.6	33.4	31.3	29.8

This comparison tends to show that valve leakage, not increased cylinder condensation, is responsible for the high steam consumptions at under loads.

Curve *A*, Fig. 3, is the flat curve, reproduced from Fig. 1 for comparison. Curve *B* represents the economy the flat-curve engine would have given if there had been 10 per cent. initial valve leakage at full load.

Pressure-plate valves, like piston valves and other types of solid valves working in holes of solid metal, wear loose instead of wearing tight, and the steam consumption increases instead of lessening. Curve *C* shows the economy of the pressure-plate valve engine with its initial valve leakage of 10 per cent. at full load, and an additional valve leakage of 10 per cent., due to wearing of valve after not more than three months' operation. Table 2 shows the method of obtaining these amounts:

TABLE 2

Valve with 10 per cent. initial leakage, and 10 per cent. future leakage:

FULL LOAD

29.8 lb. × 120 hp. = 3576 lb. per hr.
3576 + 10 per cent. leakage (or 357 lb.) = 3933 lb. per hr.
3933 ÷ 120 hp. = 32.7 lb. per hp.-hr.

THREE-QUARTER LOAD

31.3 lb. × 90 hp. = 2817 lb. per hr.
2817 + 357 lb. = 3174 lb. per hr.
3174 ÷ 90 hp. = 35.2 lb. per hp.-hr.

ONE-HALF LOAD

33.4 lb. × 60 hp. = 2004 lb. per hr.
2004 + 357 lb. leakage = 2361 lb. per hr.
2361 ÷ 60 hp. = 39.3 lb. per hp.-hr.

ONE-QUARTER LOAD

39.6 lb. × 30 hp. = 1188 lb. per hr.
1188 + 357 lb. leakage = 1545 lb. per hr.
1545 ÷ 30 hp. = 51.5 lb. per hp.-hr.

Twenty per cent. or more leakage after an engine has been in operation only three months scarcely seems credible, but a number of tests to prove this point might

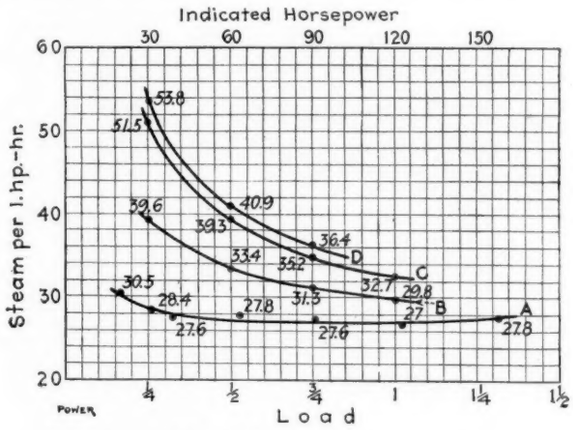


FIG. 3. A COMPARISON OF STEAM RATES

be cited: In Binghamton, N. Y., a 10x12-in. pressure-plate valve engine was replaced on the coal-saving basis by a steam-tight valve engine. The builders of the pressure-plate valve engine were allowed to place this valve in the most economical condition possible about three months before the competitive test was made.

Both engines were tested out under identically the same conditions, namely: 100 lb. steam by gage at the throttle, 1 lb. back pressure above atmosphere, and at 82 per cent. load.

The pressure-plate valve engine showed a steam consumption of 36.4 lb. per i.hp.-hr., and the steam-tight valve engine 28.3 lb., a saving of 8.1 lb. per i.hp.-hr. The valve leakage amounted to 27.5 per cent. at this load, which is equivalent to 22.2 per cent. at the full rated load of the engine.

The steam consumption per hour of the pressure-plate valve engine was 1894 lb., while that of the steam-tight valve engine was 1485 lb., or a leakage of 409 lb. per hr.,

irrespective of the load on the engine. Adding this 409 lb. steam leakage per hour to the actual hourly consumption of the steam-tight valve engine (remembering that this particular engine showed 0.7 lb. more steam per indicated horsepower per hour than the flat-curve engine) gives curve *D*, Fig. 3. In checking this curve, the fact must be borne in mind that the full-load rating of the Binghamton engine was 66 i.hp., instead of 120 i.hp., which was the full-load rating of the flat-curve engine.

As a proof that 36.4 lb. per i.hp.-hr. is not unusual for a pressure-plate valve engine, one more test on an engine of the same size and make as the Binghamton engine that had been in operation only three months, will be cited. This 10x12-in. pressure-plate valve engine showed a steam consumption of 37.8 lb. per i.hp.-hr. at full load. The engine operated under a steam pressure of 135 lb., which would tend to prove that the greater the pressure the greater the leakage past a balanced valve.

As a further proof that cylinder condensation does not increase enough to affect the economy of an engine, down to one-quarter load, tests might be cited that the author has made with superheated steam on steam-tight valve engines. The curve with superheat exactly parallels the curve with saturated steam, from full load to one-quarter load, showing that all that has been accomplished by the introduction of superheat has been to lessen the initial cylinder condensation which takes place to a greater or less extent in every engine, depending somewhat on the arrangement of valves.

Cylinder condensation, to which has been attributed the poor economy of engines at early cutoffs, should not be confused with initial cylinder condensation, which must occur with any engine using saturated steam. The former condensation is not perceptible at loads less than one-quarter, and with an initial steam pressure of 100 lb. To valve leakage not to increased cylinder condensation, should be charged the inefficiency of steam engines not employing steam-tight valves.

❋ ❋

Economy of Compression in Steam Engines

By Robert R. Fisher

SYNOPSIS—The conclusion from these tests, conducted at the Kansas State University, is that compression up to about 45 per cent. of the initial absolute steam pressure is advantageous. Above this point the economy decreases, the rate of decrease being more pronounced at light loads.

❋

Laboratory work on the economy of compression in the steam engine, so far as the author knows, has been done by only two men—the late Professor Dwelshauvers-Dery, of Liége, Brussels, and Prof. R. C. Carpenter, of Sibley College.

Professor Dwelshauvers-Dery, in 1875, carried on a series of experiments on an engine approximately 11.8x 23.6 in. (See Power, June 28, 1910.) Runs were made with compression varying from 0 to 40 per cent. Two series of tests were made: one at 45 r.p.m. and the other at 60 r.p.m. The results of these tests showed increasing steam consumption with increasing compression.

A few years ago, Prof. R. C. Carpenter conducted a series of tests at Sibley College on the high-pressure cylinder of a triple-expansion engine. (See Power, Jan. 24, 1911.) Compression was carried from 42.8 to 87.2 per cent. of the initial pressure. These tests likewise showed increasing steam consumption with increasing compression.

In the spring of 1911 a series of tests were made at the mechanical laboratories of the Kansas State University, investigating the economy of compression in the steam-engine cylinder which yielded results that throw additional light on the subject of compression. These tests were made by Ray L. Bartlett and Robert R. Fisher, senior engineers, for their graduating thesis.

Apparatus

The tests were made on the high-pressure cylinder of a compound Corliss, 10 and 24 by 30-in. engine, designed and built by the H. N. Strait Mfg. Co., of Kansas City, belted to a three-wire, direct-current generator rated at 55 kw. As the low-pressure side was disconnected, the tests can be considered as made on a simple noncondensing Corliss. The engine was designed for 200 lb. steam pressure and ran from 50 to 125 r.p.m. The inlet valves were single ported and the exhaust valves double ported. The governor was of the flyball, center-weight type, and the engine was provided with double eccentrics for both cylinders.

Manner of Conducting Tests

Before performing the tests the clearance was accurately determined by filling the clearance space with water and an investigation made to determine whether or not the valves and piston head were tight. It was found that the clearance, head end, amounted to 6.18 per cent. of the cylinder volume and at the crank end, 7.32 per cent., or an average of 6.75 per cent. Preliminary investigation showed that the piston was tight for the full length of stroke and that there was a small amount of leakage past both the steam and the exhaust valves, that past the exhaust valves being the greater.

The load was maintained by means of a water rheostat. Eleven runs of one hour each were made, with four settings on the exhaust valve to give four compression values, ranging from 21 to 92.4 per cent. of admission pressure. Runs were made at three different cutoffs for each compression. The engine would not carry full load with compression to initial pressure.

The condensate was weighed every half hour and indicator diagrams were taken every 10 min. The load was maintained constant during each run. The initial pressure at the engine was maintained constant at 115 lb. gage. A steam separator in the line just above the engine delivered steam to the engine at an average quality of 0.99.

Theoretical Considerations

Theoretically there are three points in favor of compression:

1. Compression serves as a cushion for the moving

parts and does away with the shock of the entering steam at admission.

2. It causes the cylinder to be heated up and thus reduces the initial condensation.

3. By filling the clearance space with exhaust steam, the amount of live steam entering the cylinder is reduced.

A series of calculations were made to find the theoretical steam consumption of the engine on which the tests were to be made, at four different cutoffs with four different compressions at each cutoff. In these calculations it is assumed that the expansion of steam is according to the equilateral hyberbola, for which the equation is $PV = Constant$, and that there is no interchange of heat between the steam and cylinder walls.

The results of these calculations are shown in the curves, Fig. 1. These show a decreasing steam consumption with increasing compression up to the point where

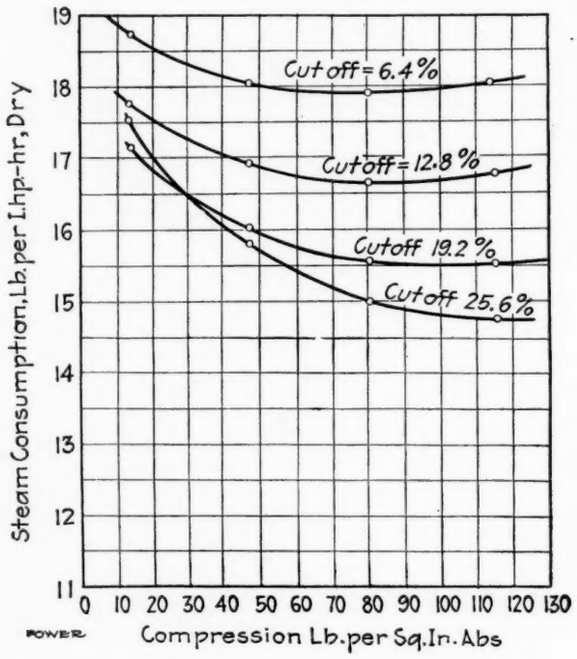


FIG. 1. THEORETICAL STEAM CONSUMPTION

the steam is compressed through a greater range than it is expanded.

RESULTS OF TESTS

The results of the tests are shown graphically in Fig. 2, as corrected for steam quality and indicator-spring error. The four sample cards in Fig. 3 show the steps by which compression was increased.

With a load of about 11 kw., the steam consumption (Fig. 2) varied within wide limits being abnormally high for both no compression and full compression. With a load of 28.5 kw. the curve, while following the same general direction, does not vary in such a pronounced manner, while the curve for 44 kw. is very much flatter.

The reasons for the gradual flattening of these curves, indicating that the effects of compression become smaller relatively as the load increases, may be seen by a consideration of the following points:

As the clearance volume is partly filled by compression steam and thus is partially nullified in effect, the steam consumption decreases until a point is reached where three other influences overcome the good effects of filling the clearance volume with steam at increased pressure. These are:

1. Condensation of the entrapped steam.

This process of using work from the flywheel to impart heat to the cylinder walls is one of low efficiency compared to the process of using the heat from initial condensation.

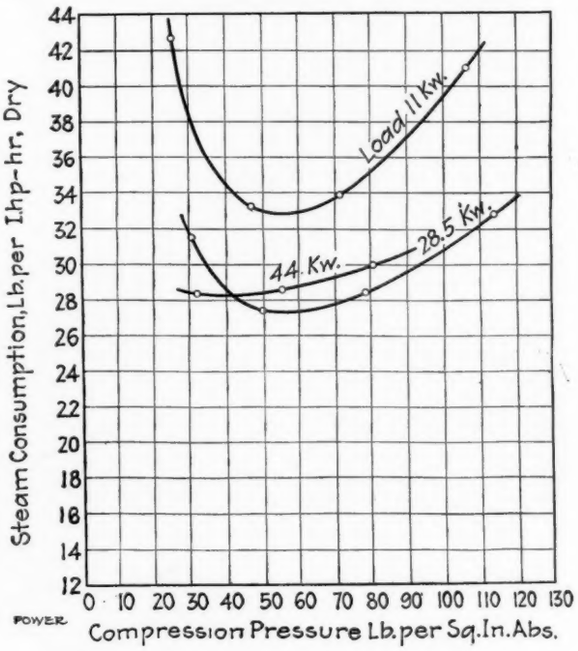


FIG. 2. ACTUAL STEAM CONSUMPTION

2. Leakage past the valves which is greater with increased pressure.

3. The compression of steam through a range approximating or greater than that through which it is expanded so that a part of the work done by the piston on the steam may be actually thrown away.

These three factors cause a rising steam consumption from this point because, while almost entirely absent at low compression, they become accentuated at high compression. When compression is carried only high enough so that condensation is moderate and compression is not used in any considerable degree as a means of heating up the cylinder walls, its action is highly beneficial in reducing the clearance volume by filling it with exhaust steam. But Professor Dwelschauvers-Dery has shown that the use of compression as a means of reducing initial condensation results only in a loss of economy.

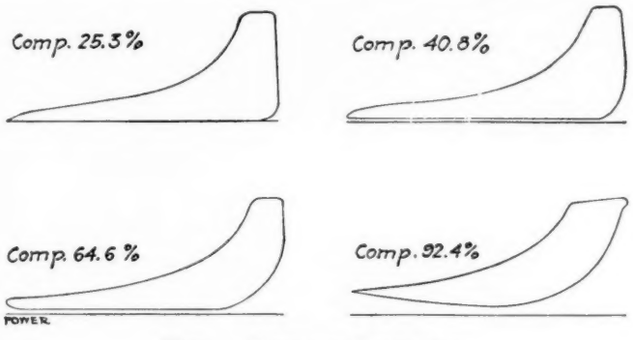


FIG. 3. TYPICAL DIAGRAMS

It may be noted here that the results of the tests made by Professor Carpenter in showing increasing steam consumption as the compression pressure was varied from 42.8 to 87.2 per cent. of the initial pressure, confirm the present results which show increasing steam consumption between these same limits. The tests made by Professor Carpenter are incomplete in not showing the effect on steam consumption of compression lower than 42 per cent.

The curves in Fig. 2 show also that clearance is detrimental to engine economy, for as the load on the engine is increased and the effect of the loss, due to clearance, becomes a smaller percentage of the steam consumption, the engine shows increased economy.

It may be argued that the cause of increasing steam consumption with increasing compression lies in the design of the exhaust valves which, not being designed for high compression, do not release as quickly as they should In reply to this criticism it may be said that it does not apply to the runs made with compression below normal. In order to see what effect this factor has at high compression the work represented by the area which would have been included had the back-pressure line been horizontal up to the beginning of compression, was calculated from a number of indicator diagrams and credited to the engine with the following results:

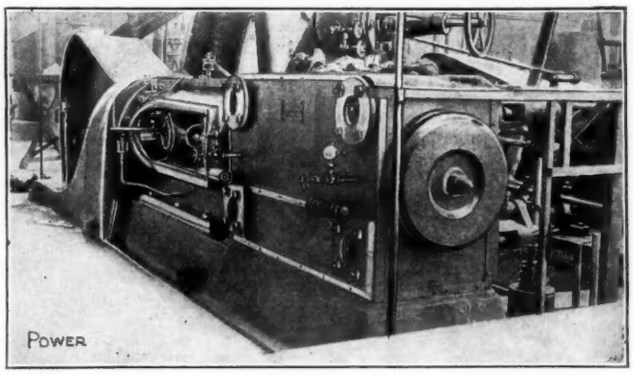


FIG. 4. ENGINE UPON WHICH TEST WAS MADE

Recalculating the steam consumption for the load of 28.5 kw., which is the only load at which the back-pressure line is noticeably abnormal, and crediting the engine with the work represented by the area above indicated, reduced the steam consumption for this compression from 32.85 lb. to 30.04 lb. per i.hp.-hr. But this still leaves the steam consumption 1.58 lb. above that for 62.6 per cent. compression.

It may be further noted that the inability of the exhaust valves to quickly release at very high compression is another point against high compression.

CONCLUSIONS

The tests show that for a 10x30-in. noncondensing Corliss with about 7 per cent. clearance and running at 112 r.p.m., compression to about 45 per cent. of the initial absolute pressure gives the highest steam economy. The exact compression, which should be carried by engines of different design running at different piston speeds, cannot be stated unreservedly, but it would seem that some degree of compression approximating 45 per cent. of the absolute initial pressure is economical for the majority of engines.

In estimating the value of these results, weight must be given to the fact that these tests were made on a commercial engine running under ordinary conditions and not on an engine especially designed for laboratory work and tested under abnormal conditions.

While leakage past the valves existed in a moderate degree, particularly past the exhaust valves, this amounted to no more than would be expected in all ordinary engines of standard design. Engineers may feel assured, therefore, that in adjusting their exhaust valves to give sufficient cushion steam to insure smoothest operation they are not sacrificing anything in steam economy but probably are adjusting for a condition that will give approximately the best results..

❋

The Wheeler Turbo Air Pump

A centrifugal pump, discharging into an ejector, will remove air and vapors from a condenser and will maintain a high vacuum, provided sufficient water at a high enough head is discharged into the ejector. But the air-entraining capacity of an ejector is limited so that such an air-removing apparatus would be costly and wasteful of power and would be unable to meet overloads due to excessive air leakage. A method of increasing air-handling capacity is to cause the water to discharge intermittently into the ejector nozzle so that it entraps air between successive discharges, as well as entraining it on its surface. A pump constructed on this principle has recently come into use in both this country and abroad.

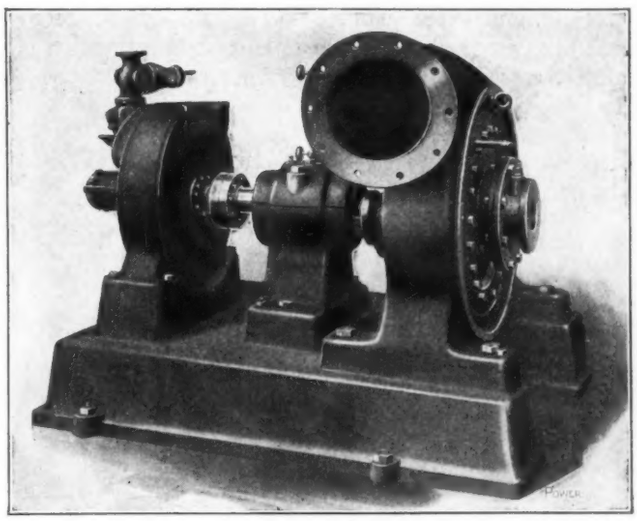


FIG. 1. WHEELER TURBO AIR PUMP

The Wheeler turbo air pump has been developed along somewhat different lines. This pump is based on the designs of the German pump built by the Allgemeine Elektricitäts Gesellschaft and is manufactured by the Wheeler Condenser & Engineering Co., Carteret, N. J. An apparent difference in design is the entire absence of the external ejector nozzle.

In general appearance the air pump, Fig. 1, is similar to a centrifugal pump. It is driven at high speed by either a motor or turbine. The pump shown diagrammatically in Fig. 2 consists of a small high-speed impeller with a large number of ports or nozzles on its periphery; secondly, a ring of tapered compression channels, and,

third, a discharge casing or volute (not shown). Hurling or throwing water enters the eye of the impeller as in an ordinary side-suction centrifugal pump and is impelled at high velocity through the ports in the impeller which rotates in the opposite direction to the impeller of a centrifugal pump. Air is admitted around the entire periphery of the impeller and is drawn into the entrance of the tapered compressor channels by the aspirating effect of the rapidly discharged water. As the slugs of water move outward they become elongated, owing to the tapering of the channels, the space between slugs decreases and the air entrapped is compressed. The mixture of air and water is discharged from the tapered channels into a volute casing, somewhat similar to that of a centrifugal pump, and is finally discharged through a suitable outlet nozzle.

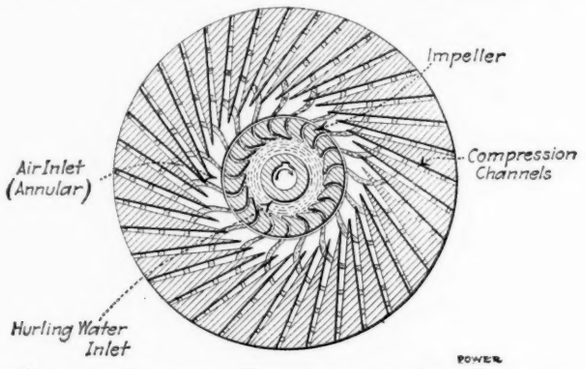


FIG. 2. SHOWING PRINCIPLE OF AIR REMOVAL

These pumps are used for both jet and surface condensers. For the latter service, the turbo air pump is frequently combined within a common casing with the condensate pump. In Fig. 3 is illustrated a combined pump of this type. Air and condensate are withdrawn from the surface condenser through a common pipe, flowing to a common suction nozzle on the pump. Just within the nozzle is a partition which separates the air and water, the latter flowing by gravity to the eye of the condensate impeller, which is of the ordinary centrifugal pump design, while the air and gases flow over the partition wall, direct to the annular air inlet surrounding the hurling water impeller of the air pump. Within the single casting two volutes are formed, one for condensate, the other for hurling water and air, so that as far as operation is concerned, the two pumps are entirely independent. However, a distinct saving results, because the amount of piping between the condenser and the pump is reduced and simplified, the space ordinarily required by the two pumps for air and condensate is greatly reduced, and both pumps are driven on a single shaft by a common turbine, thus saving power.

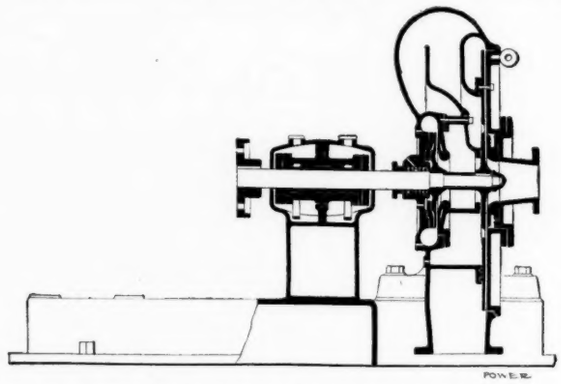


FIG. 3. CROSS-SECTION THROUGH TURBO AIR PUMP

Pumps of this type will maintain a vacuum of 98-99 per cent. of theoretical, that is within a very small fraction of an inch of the vacuum corresponding to the temperature of the hurling or throwing water. This is not only true for low temperatures of water from 32 to 60 deg., but also for hurling water at temperatures of 70, 80, 90 deg., and higher, as encountered in the summer months and in cooling-tower work. The curve of Fig. 4 shows the air capacity in cubic feet of free air at 70 deg. F., plotted against the inches of absolute pressure maintained by a recent pump. The hurling or throwing-water temperature was 78 deg. F., which corresponds to a vacuum pressure of 0.96 in. of mercury, which is to be compared with the maximum vacuum obtained by the pump of 1.1 in. absolute pressure.

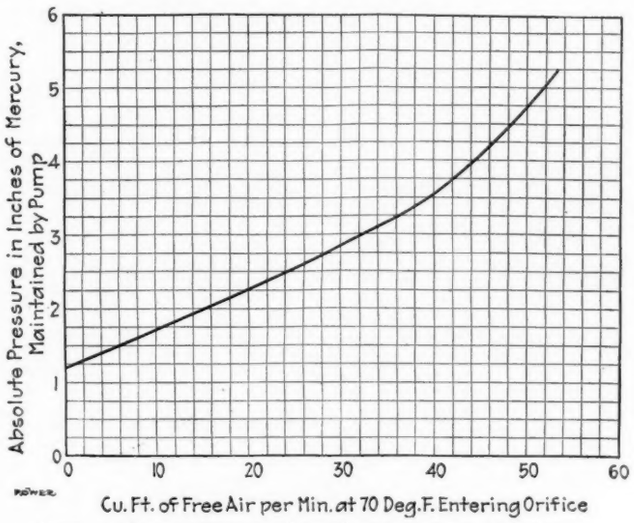


FIG. 4. RELATION OF AIR CAPACITY TO ABSOLUTE PRESSURE MAINTAINED

An important characteristic of this pump as demonstrated by this curve is the large free air-handling capacity and its ability to withstand overloads. This follows from the fact that there is a positive aspirating effect at the entrance of the numerous tapered compression channels, into which air must be drawn so long as the im-

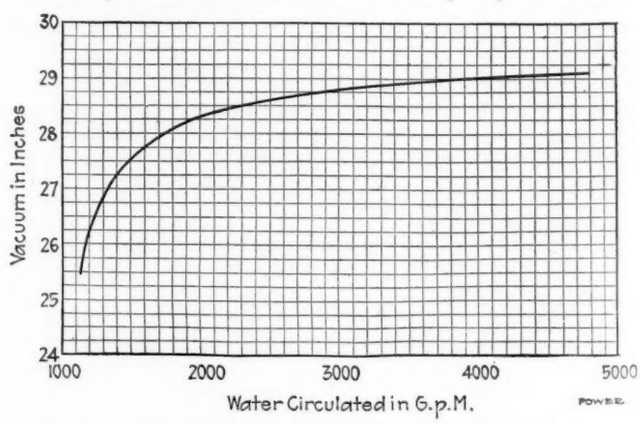


FIG. 5. RELATION OF VACUUM MAINTAINED TO QUANTITY OF CONDENSING WATER

peller rotates and discharges water. Accordingly it is found that an increase in load or the cubic feet of free air handled, does not cause failure of the pump, but simply a fall in vacuum not unsimilar to the characteristic of a displacement-type pump. The operating importance of this point is that if the air leakage on a condenser be increased at any time, due, for instance, to failure of the turbine seals, the pump will not refuse to handle the added load with a resulting break in the vacuum, but will maintain a moderate vacuum until the air-leakage trouble has been corrected. A too common experience with ejector air pumps is a failure of the vacuum because of a transient increase in air leakage.

In Fig. 5 is shown the relation between vacuum maintained by a surface condenser equipped with a turbo air pump of this type and the amount of condenser circulating water in gallons per minute. The vacuum maintained and outlet circulating-water temperatures are given for a number of points on the chart; for instance, with 4400 gal. of water per min., the vacuum maintained was 29.13 in., with the outlet cooling water at 70 deg., which corresponds to a maximum theoretical vacuum of 29.30 in.; that is, 0.17 in. higher than the actual vacuum maintained. Similarly with 1720 gal. of water per min. the vacuum maintained was 28 in. and the outlet cooling-water temperature 98.6 deg., corresponding to a maximum theoretical vacuum of 28.11 in.

Besides indicating the excellent performance of turbo air pumps of this design in connection with surface condensers, this chart is of additional interest in that it shows the increase in cooling water necessary to obtain very high vacuum. The temperature of the inlet cooling water was about 50 deg. To maintain a 29.1-in. vacuum, it was necessary to use almost four times as much water for a 26-in. vacuum and 2½ times as much water than for a 28-in. vacuum.

⁂

Power for Pumping Derived from Refuse*

Those who have not followed the more recent developments in the disposal of city refuse, will be somewhat surprised to learn that in a number of places part, if not all, of the steam used in driving steam pumps at water-works pumping stations is derived from burning waste material. As a result of the development in the incinerator method of disposing of mixed refuse in England and in continental cities, several plants have been installed in this country and, in all probability, the older and less sanitary methods hitherto in vogue will gradually give way to this more modern method, particularly as our cities increase in size.

Having produced the effect of destroying refuse in a sanitary and inoffensive manner, the question of how best to utilize this heat is answered by combining a steam boiler with the furnace and producing steam at any desired pressure. Steam so produced is available for running engines to produce electricity but, as a rule, the amount which could be produced is but a small proportion of the total amount consumed by the community; also, the operation of the destructor does not lend itself to the requirements of peak loads which are incident to all lighting and power stations. A pumping load, however, is ideal for a destructor because of its uniformity and constancy, and it is in this direction that a municipality has its greatest opportunity for conserving this energy.

It would hardly be expected that the exact amount of steam available from the destructor would coincide with the exact amount of steam required at the pumping station, but the almost invariable condition is that the destructor steam is somewhat less than required by the pumps. Therefore, the difference is made up by auxiliary boilers.

The steam may be passed direct from the destructor to the steam range of the pumping station or converted into electricity and used to drive motor-driven pumps. Instances where steam from a destructor is used directly for driving pumps are found at Montgomery, Ala.; Westmount, near Montreal, Ontario, Canada, and Savannah, Ga., while instances where the power is first converted into electricity and the current used to drive motor-driven pumps to furnish the water are found at Milwaukee, Wis., and Savannah, Ga.

*From a paper by E. H. Foster before the American Water-Works Association, at Minneapolis, Minn., June 23-28, 1913.

⁂

Peat Powder as Fuel

The announcement that Hjalmar von Porat, a Swedish engineer, had perfected a process for utilizing peat powder as fuel for locomotives has awakened interest in the possibility of developing the extensive peat bogs of Sweden for this purpose.

The powder is manufactured by the Ekelund process. It does not appear that this process has made much headway as yet, but it is now predicted that in connection with the discovery of Mr. von Porat, the use of peat powder will in time become extensive.

In the von Porat system the peat powder is fed by an automatic device into the furnace of the locomotive, which is specially arranged to consume it. According to Mr. von Porat, the results obtained with peat powder may be summed up as follows:

Substantially the same results can be had from 1½ tons of peat powder that 1 ton of coal will produce. Peat powder may be burned with an admixture of about 5 per cent. of coal. As to firing with peat powder, the work is almost nothing in comparison with firing with coal, because the powder is forced into the furnace by automatic process. No change had to be made in the boiler and none in the fire box, except installing the special apparatus. There is no difficulty in bringing the powder from the tender to the fire box, as it passes through a conveyance pipe. Another advantage in using peat powder is that no cold air can get into the fire box and neither smoke nor sparks escape from the smokestack.

As a result of von Porat's invention, it is reported that a number of the Swedish railways are preparing to use peat powder instead of coal.

⁂

It is reported that a compressor which has been running at the power house of the Nevada Consolidated Copper Co., 24 hr. per day for the last five years is compressing 16,000,000 cu.ft. of free air per gallon of lubricating oil and without bad effects on the machine. This is an excellent record and is worth looking into a little. A compressor with an intake cylinder 24 in. in diameter, which is quite a common size, and a piston speed of 400 ft. per min., will compress, say, 1200 cu.ft. of free air per minute, or 1,728,000 cu.ft. per 24 hr. Then the oil used per day, at the rate of consumption given above, would be only 0.108 gal, or less than a pint for the 24 hr.—"Compressed Air Magazine."

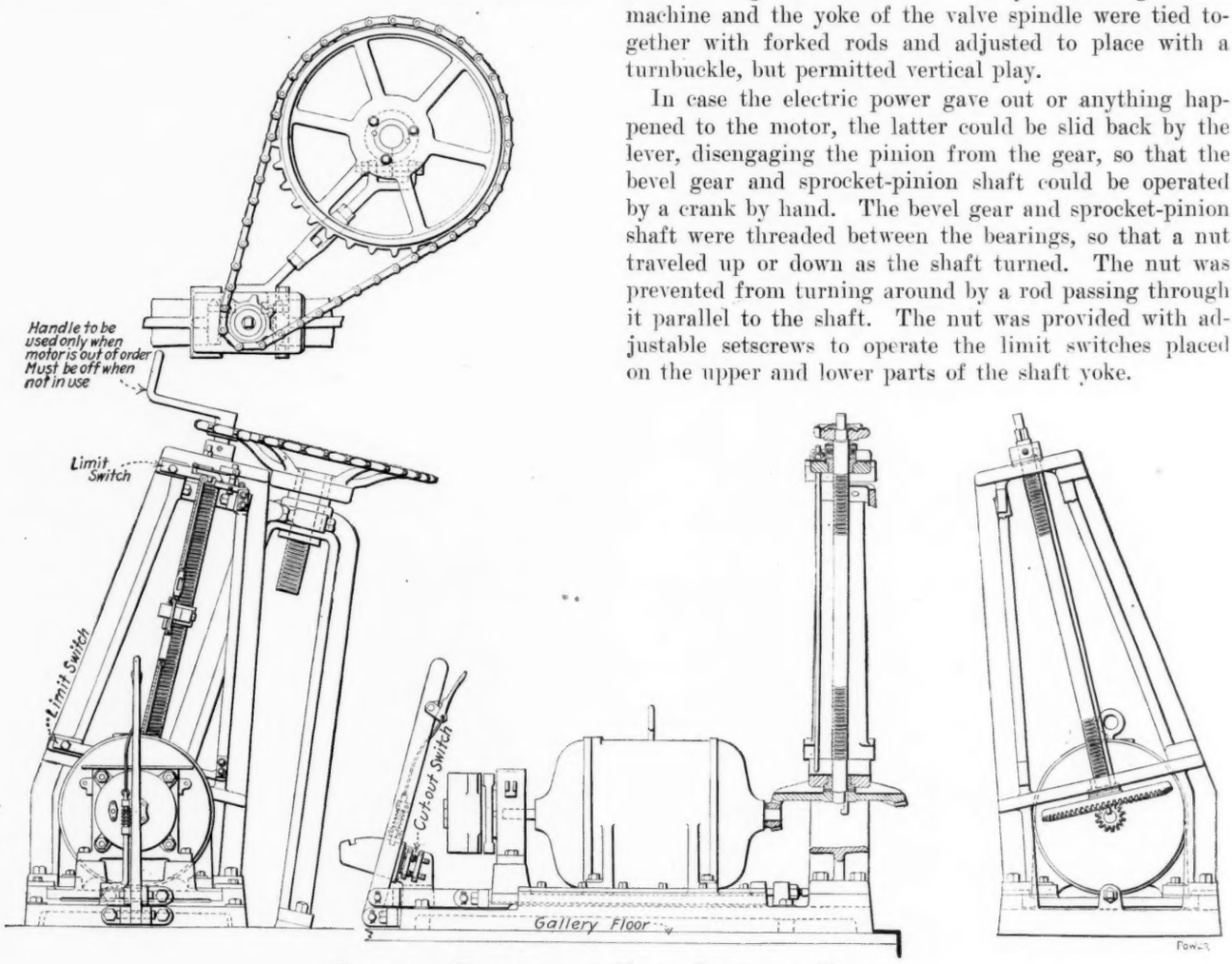
ELECTRICAL DEPARTMENT

Motor Drive for a 28-in. Valve

By James R. Allen

Some time ago the writer designed a motor drive for a 28-in. exhaust valve which proved very satisfactory. The turbine was situated on the gallery floor of the power house and exhausted through a 28-in. crane valve to a jet condenser which rested on the basement floor and stood up in the opening between the galleries. The exhaust-valve stem came up just outside the gallery floor and tipped one hole to clear the side of the condenser so that to operate the valve a man had to lean over the gallery railing and turn the valve 57½ times to open or close it, which took too much time and labor.

Because of the elevation of the gallery floor and the variation of the center line of the 28-in. exhaust valve, when the condenser was filled with water a chain and sprocket drive, which permitted flexibility, was used. The device was set opposite the valve close to the edge of the gallery floor.


Handle to be used only when motor is out of order Must be off when not in use
Limit Switch
Limit Switch
Cut-out Switch
Gallery Floor

Plan and Elevation of Motor Drive for Valve

An old 2-hp., direct-current motor was found and set on the base, sliding in ways on the bedplate. An electric brake was also connected to the motor shaft and fastened to the base so that when the current was thrown off the motor it would stop instantly.

The yoke casting, carrying the bevel gear and sprocket-pinion shaft, was fastened to the bedplate and tipped to suit the angle of the valve. The yoke casting of the machine and the yoke of the valve spindle were tied together with forked rods and adjusted to place with a turnbuckle, but permitted vertical play.

In case the electric power gave out or anything happened to the motor, the latter could be slid back by the lever, disengaging the pinion from the gear, so that the bevel gear and sprocket-pinion shaft could be operated by a crank by hand. The bevel gear and sprocket-pinion shaft were threaded between the bearings, so that a nut traveled up or down as the shaft turned. The nut was prevented from turning around by a rod passing through it parallel to the shaft. The nut was provided with adjustable setscrews to operate the limit switches placed on the upper and lower parts of the shaft yoke.

The limit switches were set so that the current would be shut off just before the valve seated itself or was fully opened. The switches were provided with carbons to draw the arc so as not to burn them. The upper and lower hinged parts of the limit switches were connected by rods and a tension spring so that they were always held on their contact bars unless lifted by the traveling nut and the contact broken, thus stopping the motor.

A cutout switch connected before the motor with its terminals fastened to the lever sector and a contact bar on the lever, prevented the motor destroying itself if it was disengaged from the bevel gear and the current was

accidentally thrown on. A sprocket wheel was fitted to the nut on the valve spindle in place of the handwheel and connected with a sprocket chain to the sprocket pinion.

The bevel pinion had 16 teeth, the bevel gear 80 teeth and the traveling nut was threaded 11 threads per inch with approximately 30-in. travel. The sprocket pinion had seven teeth, and the sprocket gear 34 teeth, so that the valve-spindle sprocket gear made 74 r.p.m. with the motor running 1800 r.p.m.

A forward and reverse switch was placed near the turbine throttle so that the engineer could open or close the valve at pleasure.

The device did not cost as much to make as to purchase a regular motor-operated valve. It is practically foolproof and has not been out of order since it was made.

Some Electrical Fallacies

By Arthur H. Allen

In all vocations there are certain beliefs more or less firmly rooted in the popular imagination, but actually based upon erroneous ideas, and the electrical industry is no exception; indeed, in earlier days it bristled with fallacies, but many of these have been exploded. Nevertheless, there is still a fair number which persist in spite of the advance of technical education and of repeated disproof.

The writer recently heard it stated that a motor could be built which would run faster if the load on the shaft were increased. A similar statement, many years ago, induced him to try experiments on various motors, changing the excitation, the position of the brushes, and so on, with a view to putting the statement to test, as he doubted the possibility of such an achievement. The results were entirely negative, and consideration shows that this must necessarily be so. In the case of a direct-current motor, the current through the armature (at constant voltage) cannot possibly increase unless the counter electromotive force is first reduced, and this cannot take place unless either the speed is lowered or the field weakened. But the armature cannot carry an increased load unless the current through it is increased, or the field strengthened. With constant voltage, the latter alternative is inadmissible; therefore the speed must fall when the load is increased. A larger current then flows, and this may weaken the field in two ways: by armature reaction, or by a series winding in opposition to the shunt winding, or both. In order to restore the original speed, the field must be weaker, while on the other hand, in order to carry the increased torque, the product of armature current and field strength must be greater than before Armature reaction alone is not sufficient to bring about the necessary weakening of the field, so that a shunt motor always falls in speed as its load increases. By means of a differential series winding, however, it is possible to weaken the field when the current increases, and thus the speed can be maintained very nearly constant.

The action is similar to that of a mechanical governor on a steam engine, which can be so designed and adjusted that a very small fall in speed results in the admission of an increased quantity of steam at each stroke to carry an increased load; but it is well known to engineers that if the governor is made too sensitive the engine will hunt—the correction applied through the action of the governor may be too great, and the speed of the engine will become irregular. Hence it is necessary to allow a drop in speed of at least 2 or 3 per cent. between no load and full load, to insure stable running, and precisely the same thing applies to the direct-current motor. If the compensation for increased load provided by the differential series winding is too great, the speed will become unstable. Hence, it is necessary to allow a small but definite drop in speed with increasing load. With the aid of relays, the problem can be solved, but that is another question.

Another fallacy which is still often met with is that of "bedding" in the winding of a solenoid. It is stated that some economy in winding space can be derived from the bedding of round wires in the successive layers, as shown in Fig. 1, but this is a wholly erroneous view. The successive layers are respectively right- and left-hand spirals, and if one considers the analogous case of a left-handed bolt screwed into a right-hand nut, it will at once be seen that the respective threads cross one another, so that the bolt cannot even enter the nut unless the outside diameter of the former is equal to or less than the inside diameter of the latter. They will not go together at all. The threads of a right-hand bolt and a right-hand nut, however, if of the same pitch, "bed" together perfectly. Now, whether the cross-section of the former on which a

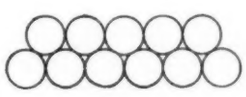


Fig. 1.

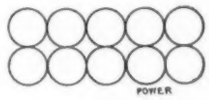


Fig. 2

coil of wire is wound be round, oval, or rectangular, the fact remains that the windings are alternately right- and left-hand spirals, and even if they bed on the corners of the former, the turns must cross one another at two points in each convolution; at those points the layers are in the position shown in Fig. 2, and there is no bedding.

Again, take the "extra current," so called, which flows at the moment of interrupting an inductive circuit, such as that of a field-magnet coil. There are many who believe that at that moment there is a rush of current, of greater amperage than the current which was flowing before the interruption commenced. The name certainly suggests this idea, which, however, is totally wrong. Of course, the current persists for a short interval, producing a spark; but the act of interrupting the circuit is simply that of increasing the resistance in the circuit more or less abruptly, and the current actually diminishes from the first moment. What really happens is that the fall in the value of the current, which is necessarily accompanied by the collapse of a number of lines of force (using the ordinary convention), and, therefore, by the intersection of the convolutions of the winding by lines of force, results in the production of an electromotive force acting in the same direction as the original electromotive force, and thus tends to maintain the current flowing, or, in other words, which, perhaps, more correctly describe the phenomenon, it opposses the change in the conditions, in accordance with the law of Lenz. If the circuit is very suddenly interrupted, this induced electromotive force

may reach an extraordinarily high value, and has often been known to cause the puncture of the insulation of a solenoid; it is, indeed, an "extra voltage" in the sense of an increased voltage, but *the current diminishes all the time.*

Take now the question of overleaping of current when switching on a tungsten lamp. The resistance of the filament is low at first, the filament being cold. A given voltage, therefore, produces a current far in excess of the normal current, causing the filament to be heated very rapidly. It is easy to fancy that the heating has a sort of inertia, which causes it to overleap the proper limit, so that the filament attains a higher temperature than the normal ultimate value, but this is an erroneous view. The current is always equal to the ratio of the effective voltage in the circuit to the resistance in the circuit at the moment, and as the temperature of the filament (and, therefore, its resistance) rises, so the current necessarily falls, until the point is reached where the current flowing just suffices to maintain the temperature that the filament has attained. A good analogy is a tank fitted with an automatic ball-valve connected with a water supply. If the water is turned on when the tank is empty, it rushes in at first at full bore; then as the tank gradually fills, the ball rises and gradually closes the valve, till when the tank is nearly full the water comes in at a mere trickle. When the tank is quite full the flow of water has been completely stopped. The tank is never filled above that limit. There is only one way in which overfilling could be produced—namely, by the presence of inertia of some kind in some part of the apparatus.

In the case of the tungsten lamp, the electromagnetic inertia known as inductance may supply that condition. Another factor by which the effect of inertia may be produced is that of capacity for heat; the parts of the filament close to the terminal electrodes, and those which are in contact with the supporting hooks, do not heat up so rapidly as the intermediate parts, and thus their resistance does not increase at the same rate. The result is that the resistance is less, and, therefore, the current is greater, at the moment when the temperature of the filament passes through the normal limit, than they will ultimately become, and this would cause a momentary excess of heat and brilliancy. That such overleaping does take place in many cases cannot be denied; but it is not due to the upward rush of the temperature, but to causes such as those above mentioned. No circuit is completely noninductive, and no tungsten lamp of ordinary dimensions is free from the wire hooks, etc., which might account for the phenomenon.

Lightning Protection*

There are two classes of lightning strokes, direct and indirect. The direct stroke is, as the name implies, something in the direct path of lightning to the ground. This kind of a stroke is so powerful as to render all present forms of lightning protection useless. A power wire exposed to this severe condition is most likely to be exposed to a power arc which follows lightning to the ground and the arc often burns the line conductor in two. The means of support for the power wire sometimes aid or add a destructive element to the whims of lightning. Wooden pole lines are often interrupted because of a shattered pole, and in few instances does lightning strike the wires midway between supports. Again, more trouble has been evident on wooden-pole lines than upon tower lines. This is because of the greater resistance to the flow of lightning offered by the wooden-pole line, whereas the steel towers are of ground potential.

The indirect stroke of lightning is detrimental to good service because of the effect produced. This induced stroke which affects the line in the immediate vicinity of the direct stroke, builds up a potential nearest the path of the lightning often of such magnitude as to strike out and attract the lightning to the conductor.

It would seem evident that bringing the potential of the earth to an elevation higher than the power wires would be the best means of protection. The most usual manner of obtaining this protection, in a degree, is to place a single wire at the extreme top of the structure and have this parallel the main power conductors. This conductor is grounded at intervals, and is known as the ground wire. When lightning is being conducted to earth along this wire, an induced potential is often produced on the main-line conductors, occasionally so high as to cause a "spill-over" of the insulators. To prevent this dangerous rise of potential on the line, it is best to ground the wire about every 375 ft.

The multigap arrester, at the time it was devised, afforded what was considered the best means of protection. It had the disadvantage of having parts affected by the current which, in turn, affected the sensitiveness of the arrester. That portion affected was the gnarled edges which burned off and increased the gap, thereby making it necessary to attain a greater potential before a path to the ground is obtained. This arrester is in use on voltages up to 33,000 and 44,000 volts.

An arrester known as the compression type has been produced for voltages up to 10,000. This is a form of multigap arrester embodying similar principles with distinct feature. It requires little space, and is mounted on the transformer pole, one arrester placed in each phase. Upon a short-circuit it attains a temperature such as to cause a cap to be forced off, and in this way opens the circuit.

The open-air horn-gap arrester is a combination of a horn gap and a choke coil; the action is as follows: The surge passes along the line to a choke coil, and it is immediately retarded until a potential great enough to jump the gap is attained, at which time the choke coil acts as a magnetic blowout, and the arc is carried up the horns and is dissipated. It is recommended to place a fused horn gap in series with the line to prevent heavy rushes of current. The fuse prevents its being a self-sustained arrester.

The electrolytic arrester is still the arrester upon which the greatest burden of protection lies. This has remained of the same design and has proved to be the most satisfactory means of protection from lightning on systems of high voltage. It dissipates surges from the line, and removes abnormal conditions.

In any case, the arrester used depends primarily upon the ground connections. One of the best methods of obtaining a ground in a station is to drive galvanized-iron pipes into the earth at different places, all the pipes to be well salted just below the surface of the earth and electrically tied together.

*Excerpt from report of the Electrical Transmission Committee of the Ohio Electric Light Association.

GAS POWER DEPARTMENT

Effect of Variable Load on Gas Producers

By Reginald Trautschold

In the past manufacturers of suction-gas producers often made claims for economic operation that were not realized in ordinary practice, although they may have been exceeded in tests made at the factories or in certain installations where all conditions were peculiarly favorable. Much progress has now been made in the design of gas producers and the conservative manufacturer no longer guarantees exaggerated economies, such as a horsepower for ¾ or 1 lb. of fuel, it being possible to get only a certain number of B.t.u. from a fuel and it is in but rare exceptions that more than one-third of the available heat can be converted into useful energy. At present, the rating of gas producers is much more liberal than formerly and few manufacturers now care to guarantee a horsepower on less than about 1¼ lb. of anthracite coal, although some installations are doing better than this.

The manufacturer is now confronted by another trying proposition, namely, to guarantee the effect of variable load on the economy of the gas producer. Possibly this is another stumbling block, for guarantees such as a fuel consumption of 1¼ lb. per b.hp.-hr. from three-quarters to full load, and 1½ lb. per b.hp.-hr. with half load, are made by some of the most conservative manufacturers. Owners should know that a gas engine is not equally efficient under all degrees of load and, therefore, should not expect to be protected by any such guarantee on the part of the producer manufacturer.

Variable load has little effect upon the economy of a correctly designed gas producer; which, instead, depends directly upon the efficiency of the gas engine, and any guarantee that the fuel consumption of a gas producer will not exceed a certain amount under a given load must be extremely conservative to be realized in every installation. The only reliable guarantee that can be made is that the fuel consumption of the producer will not vary more than 2 per cent., for any load, from the efficiency of the power unit operating under the same load. The efficiency of a gas engine varies under different degrees of load and so many conditions enter into the problem that it may be impossible for the gas producer to be operated efficiently because of inefficient operation of the gas engine. It is possible, of course, to guarantee a certain rate of gas generation in a suction-gas producer from a particular grade of coal, but such rating would not be convenient nor satisfy the operator. He desires to know how much fuel is required to generate certain power at the engine.

A suction-gas producer consists simply of a closed chamber in which a solid fuel is converted into a combustible gas. This conversion is only possible through the admittance to the producer of a certain quantity of air for combustion and the reducing medium that brings about the chemical reduction in the products of combustion. The supply of these direct and negative reaction mediums depends upon the suction of the engine, and, hence is directly dependent upon the engine requirements for fuel. This withdrawal of gas is equal to the consumption of fuel by the engine and it is, therefore, impossible to have other than the same economy in both the producer and the engine, with possibly the exception of the 2 per cent. referred to in the suggested guarantee. This 2 per cent. may occasionally occur, due to the fact that a gas producer is very similar to a thermos bottle, storing up heat, so that the fire bed is maintained at a practically constant temperature, irrespective of the rate of gasification in the producer. A small portion of any producer gas consists of hydrocarbons that are driven off from the fuel; therefore, a slight excess of such hydrocarbons might be present in the delivered gas at a time of supplying green fuel, thus increasing the richness of the gas during a short period. A varying richness of gas reduces the efficiency of the gas engine—a reduction that can rightfully be charged against the gas producer.

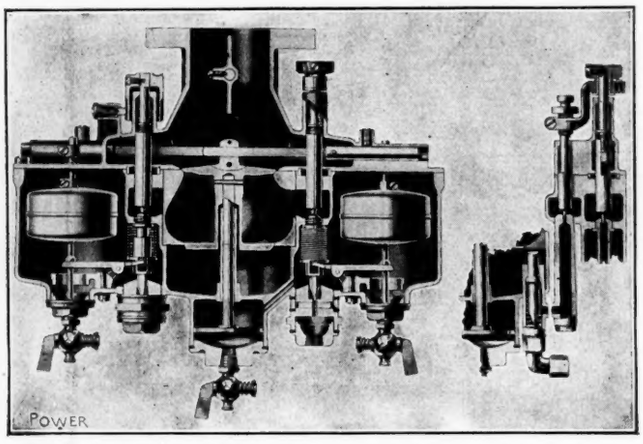
Sudden variations of load are conducive to reduction of engine efficiency, naturally, and correspondingly to poorer economy of fuel consumption by the producer, for no engine governor has yet been devised that can instantly take care of a considerable drop or increase in load. The period of reduced efficiency of the engine is ordinarily of sufficient duration for the gasification in a gas producer to accommodate itself to the demands for fuel, and several of the most successful gas producers depend for their economic operation on this fact—a dependence that actual operating results indicate as permissible. One eminently successful producer, however, is provided with an additional precaution against unnecessary gasification through a bypass connection leading from the gas holder back to the scrubber. In this producer, excess of gas, at a time of sudden drop in the engine load, returns to the scrubber and thus the vacuum throughout the gas-producer system is reduced and gasification is retarded until the engine load increases and the demands for fuel are correspondingly greater.

Standby periods represent certain losses in efficiency of all generators, but here the gas producer is far more economical than steam boilers. Gasification practically stops on the breaking of the vacuum in the system—on a shutdown. That is, on the cessation of operation, the purge or stack pipes of gas producers are opened to the atmosphere and the complete combustion of fuel is necessarily very slow. The condition existing in the producer is a relatively deep mass of incandescent fuel with very little oxygen supply for combustion. The ordinary air supply for the fuel charge is practically cut off and little air enters the producer chamber, except through leakage or down the open stacks. Combustion being thus minimized and there being little circulation, the producer stores up its heat during inoperative hours so that when operations are again resumed by the admittance of air,

the fuel bed is easily revived. The dormant condition of the producer can be maintained for a considerable period, ample to care for any ordinary standby condition. Even should such a period extend over a week or two, it is uneconomical to pull the fires. During a standby period of such length it is customary to admit no air during the first 12 hr. or so and then only sufficient air to revive the fire somewhat, when it should be again cut off and the producer allowed to remain in the dormant state for a day or two before again reviving the fuel bed. Charging the producer or supplying it with fuel during lay-over periods is also inadvisable, for such new fuel tends to bake or fuse together, increasing the difficulty in starting up the producer. Should the producer-fuel charge, at time of shutdown, be insufficient to last over the standby period it is more economical to pull the fires than to attempt to maintain them by the addition of fuel. Starting up a producer after a shutdown over night or after a lay-over period necessitates some waste of fuel during the time required to revive the charge by "blowing"—ordinarily not more than 15 or 20 min.—but this loss is practically the only direct loss of the gas producer.

New Combination Carburetor

Charles Rayfield, inventor of the well known Rayfield carburetor used on automobiles, has been experimenting for some time on kerosene carburetion, and has brought out a combination carburetor employing kerosene, gasoline and water, which is intended for use on traction, stationary and marine engines.



SECTION THROUGH RAYFIELD CARBURETOR

The operation of this carburetor is automatic. It is so designed that a proper mixture is supplied under all speeds and under all loads, using kerosene, distillate or other low-grade fuel. The fuel and air openings are operated by a floating double-poppet valve, which is lifted by the suction of the motor in proportion to the speed.

Two float chambers are used, making it possible to start at any time on gasoline. The shifting of a small lever by the operator shuts off the gasoline and opens the kerosene port, instantly changing the mixture from one fuel to the other.

Separate adjustments are necessary for each fuel and this requirement is met by the installation of separate needles, thus making it unnecessary to change the adjustment when switching suddenly from one fuel to the other. One air inlet answers when using either fuel. Heat to assist in vaporizing the fuel is supplied by a hot-air tube from the exhaust manifold.

This carburetor is not intended for use on automobile motors, but it is understood that Mr. Rayfield is now experimenting along this line. The carburetor is made by the Findeisen & Kropf Manufacturing Co., Chicago.

Splash Lubrication

One noticeable feature in the operation of gas engines is that the vertical engine running with the splash system consumes more oil than the hand-oiled horizontal engine of the same horsepower. In the case of the hand-oiled engine, if the cups are feeding too fast they can easily be adjusted, but in the case of the splash system one must keep the oil to a certain level in the crank case and let the connecting-rod bolts throw all the oil that will cling to them. It is common to find a 100-hp. splash-feed engine consuming 4 to 8 gal. of oil in 24 hr.; while 1 gal. will easily lubricate the hand-oiled engine.

I had a case some time ago of a vertical engine that consumed 5 gal. in 11 hr. A representative of a well known oil firm came along and said if I would use his oil it would only take 1 gal. per day as the oil I was using was no good. This new oil cost three times as much as the old. However, after considerable controversy, our firm decided to order two barrels. The crank case was cleaned out and 35 gal. of the new oil was put in and in order to keep the proper level while the engine was running it was necessary to use just as much of the high-priced oil as of the cheaper oil. I decided to lower the oil in the case and stop the engine throwing so much up in the cylinder. Therefore, I lowered the level so that the bolts passing through the end of the connecting-rod and through the crankpin bearings did not touch the oil.

I took out a bolt at the back of each connecting-rod and drilled and tapped a hole in the lower end for a ½-in. stud; then bent the stud to a right angle at the bottom after having flattened the end to a spoon shape. I screwed the stud into the end of the bolt and secured it with a locknut, then replaced the bolt. The stud dipped in the oil to a depth of 1¼ in. When the engine was started, I took care to see that the cylinders were not running hot and after a 5-hr. run shut down and found the cylinder walls well oiled. This cut the oil consumption down to 1 gal. per day. The 7x18-in. bearing, however, was not getting enough oil, so something had to be done to use the oil over again from the case. I, therefore, procured a small plunger pump which had been used to circulate the jacket water on an automobile; it was 1 in. diameter by 1¾ in. stroke. This I attached to the side of the crank case and drove it by a ⅝-in. connecting-rod from the exhaust camshaft. The pump discharged into a 2-gal. tank situated on the top of the crank case between the two cylinders from which the oil was piped to run through two large (quart) sight-feed oil cups, one on each outside main bearing; also a pipe led to the center bearing in the crank case. Thus the oil ran in a steady stream all the time the engine was in motion. This method supplied the bearings with about 10 gal. of oil an hour, whereas before they only received 1 gal. in 10 hr.

This not only reduced the wear on the bearings, but cut down the oil bill $300 in a year.

Stratford, Ont., Can. HARRY WESTWOOD.

HEATING AND VENTILATION

High-Pressure Steam Heating vs. Hot-Water By-Product Plant

By Ira N. Evans

In the issue of Power, June 3, 1913, is a reprint of a paper by F. W. Ballard entitled, "District Steam Heating with High-Pressure Steam."

In the paper above referred to hot water is not considered, but the data of the problem will be used as a basis to determine the financial possibilities of a vacuum hot-water heating system, where turbines could be operated under partial vacuum and the latent heat of the steam utilized in its passage from the engine to the condenser.

With the long distances and consequent expense of mains it is a close proposition from a financial standpoint for any system, and the use of exhaust steam directly in the mains would be out of the question financially. From the comparison as stated with 40 lb. of steam per kw.-hr. for heating and 15 lb. per kw.-hr. for full vacuum, it would seem impossible for any byproduct plant to be successful even for short distances.

In an article in Power, Nov. 26, 1912, on the "Cost of Exhaust-Steam Heating," methods are discussed showing that by the use of hot water under forced circulation for heating, the steam rate for power may be reduced by lowering the temperatures at which the exhaust steam is utilized. At the same time the radiation is operated in extreme weather at the same temperatures as vacuum steam. Under the conditions named the steam rate of the high-vacuum turbine will not be increased to so great an extent and a broader field is opened for byproduct district-heating plants where they would not pay at present.

The total steam for 1000-hr. operation for the proper percentages of time for each outside temperature was given in the article as 59,782,363 lb., or an average of 59,782 lb. per hr. With a maximum rate for heating at —15 deg. outside temperature of 113,832 lb. per hr., the average quantity of steam would be 52.52 per cent. of the maximum. The total kilowatt-hours, including 200,000 for pumping, was 2,090,928, which would give an average rate of

$$\frac{59,782,363}{2,090,928} = 28.6 \textit{ lb. per kw.-hr.}$$

for heating and power. The steam rate at 28 in. of vacuum was 20 lb. per kw.-hr., on a 500-kw. unit, and with the allowance for the difference in feed requirements at 28 in. of vacuum, or 100 deg. and atmosphere or 212 deg., the rate increased to 22.24 lb. per kw.-hr. from and at 212 deg. The average increase in steam consumption if the 15 lb. in Mr. Ballard's paper included the auxiliaries would be

$$\frac{28.6}{22.25} \times 15 = 19.2$$

or, say, 20 lb. per kw.-hr. for the average rate for heating. The 15 lb. per kw.-hr. is low, but the relation cited in the author's article having been proved out in actual practice, should hold for any other like condition.

An increase in the steam rate from 15 to 20 lb. or 33 per cent., as against 15 lb. and 40 lb. per kw.-hr. changes the complexion of the problem. In Mr. Ballard's paper the full load requires a 15-lb. drop in pressure for the low-pressure system and 68-lb. drop in the high-pressure system. This condition will occur about 2 per cent. of the time, although it has to be provided for in boiler capacity. If the loss in pressure varied as the square of the weight of steam the pressure drop under average conditions with 60 per cent. load factor would reduce to

$$0.60 \times 0.60 = 0.36 = 36 \textit{ per cent.}$$

Then the average drop on the low-pressure system would be

$$0.36 \times 15 = 5.4 \textit{ lb.}$$

and the average drop on the high-pressure system

$$0.36 \times 68 = 24.48 \textit{ lb.}$$

Even with 5.4 lb. back pressure it would be difficult to obtain a rate of 40 lb. per kw.-hr. and vacuum on the end of the line would increase the steam volume, requiring enlarged mains and increased expense of installation.

For loss in the value of the exhaust steam for heating, due to its passage through the engine, Mr. Ballard deducts 20 per cent. If turbines are used the exhaust steam should be slightly superheated or at least dry for the same reason that dry or slightly superheated steam is obtained at the end of the heating line. The reciprocating engine may or may not give dry steam, but if cylinder condensation is reduced to a minimum and superheated steam is used, as would of necessity be required for a rate as low as 15 lb., the exhaust steam would be nearly dry at the engine, and there would be no occasion for deducting so large a quantity as 20 per cent.

In comparing a byproduct plant with a straight heating plant the boiler power under average conditions will be 50 to 60 per cent. of that required under maximum conditions. The main saving in a byproduct plant is in the investment in buildings, boilers and equipment required for the maximum load over the average load. The spare boilers for both power and heating can be reduced and made to serve two purposes.

This is one of the main objections to the use of hot-water boilers for district-heating hot-water plants. The extra labor and the fact that the heating plant is discontinued in summer while current is required the year round are two additional arguments in favor of a byproduct plant. With regard to the statement that the loads on the heating and power do not come at the same time, this is largely a matter of business management in obtaining a proper power load to balance the heating. This is one of the prime requisites for a successful byproduct district-heating plant.

With the hot-water plant already referred to in this discussion, practically the same power load in kilowatts would be carried throughout the heating season. The heating requirements are met by simply varying the vac-

uum and steam temperatures with nearly the same power load and increasing and decreasing the steam rate. The consumption would be 30 lb. or more in zero weather and 16 or 17 lb. at 26 in. of vacuum and 55 to 65 deg. outside temperature. This has all been demonstrated in actual practice and all that is necessary is a constant power load sufficient to balance the heating. All surplus or peak would be carried on a separate machine under full vacuum.

The hot-water layout provides a single main in each street with the same capacity to the points indicated in Mr. Ballard's paper. This requires an 18-in. supply main to Euclid Ave. through Wilbur Ave. and One Hundred and Fifth St., a 12-in. return main through One Hundred and Seventh St. to Fairmount Road and Cedar Ave., and a 14-in. O.D. main along Euclid, Carnegie and Cedar. This line and the 12-in. main from One Hundred and Seventh St. join into an 18-in. return main leading back to the plant. The expensive reducing valves are not required on this layout, and any live steam required will be utilized under boiler pressure and temperature in a separate live-steam heater with gravity return to the boilers. The exhaust steam would be utilized in a separate heater or condenser and all condensation would be saved.

A 25-deg. drop and 75,000,000 B.t.u. per hr. as a maximum heating load would require 50,000 lb. of water to be circulated per minute; 30,000 lb. through the 12-in. main and 20,000 lb. through the 14-in. O.D. main. There are 7000 ft. of 18-in. main, 2700 ft. of 12-in. main and 8000 ft. of 14-in. O.D. main, making 17,700 ft. of main as compared to 8200 ft. on the steam system. This arrangement will require a 225-ft. head on the pump and house connections will be shunted off the main with sufficient distance between the terminals to produce circulation. The power to drive the pumps will be

$$225 \times 50{,}000 = 11{,}250{,}000 \textit{ ft.-lb. per min.}$$

or 341 theoretical horsepower. At 70 per cent. efficiency 487 hp., or 365 kw., will be required to circulate the water. It has been demonstrated in former articles that nothing is lost in the pumping. The heat units developing power are all returned in friction and the heat in the exhaust steam is utilized for heating the water. The pumping, however, represents compulsory live-steam operation and none of this heat is available for power.

It is assumed that the 15 lb. per kw.-hr. under 28 in. vacuum is obtained with steam at 150 lb. gage and 100 deg. superheat, and that the average temperature of the heating medium is 150 deg., permitting a vacuum of 20 in. and a temperature of 160 deg. for an average outside temperature, of, say, 32 deg.

The data for loss in the mains for the three systems are given in Tables 1, 2 and 3. The water system has an average assumed temperature of 160 deg., although the actual variation would be from 100 to 215 deg., depending on the outside temperature.

Mr. Ballard mentions the nonconductivity of the ground, although the efficiency of the covering is almost what is found when the pipes are covered and surrounded by air. This is made up in part by the high specific heat and density of the surrounding earth. High absorbing power is nearly as great a source of loss as great conductivity, although constant heating will eventually raise the ground temperature and reduce the loss.

The tables show under the assumed conditions the line loss on the high-pressure steam system to be 1,749,982 B.t.u. per hr., on the low-pressure steam system 2,491,776 B.t.u. and on the hot-water 2,341,418 B.t.u. per hr. The high-pressure steam system will have a nearly con-

TABLE I. RADIATION FROM MAINS OF LOW-PRESSURE STEAM SYSTEM

Size In.	Sq.Ft. per Ft.	Length Ft.	Sq.Ft. Surface	Steam Temp. Deg. F.	Steam Temp. 40 Deg.	B.t.u. per hr. per Sq.Ft.	B.t.u. per hr. Bare Pipe	15 per cent. Loss B.t.u.
24	6.238	2000	12,476	247	207	414	5,165,064	774,760
22	5.76	2000	11,520	241	201	402	4,631,040	694,656
18	4.71	2200	10,362	236	196	392	4,061,904	609,286
14	3.662	2000	7,324	228	188	376	2,753,824	413,074
Total								2,491,776*

* 2492 lb. steam.

TABLE II. RADIATION FROM MAINS OF HIGH-PRESSURE STEAM SYSTEM

Size In.	Sq.Ft. per Ft.	Length Ft.	Sq.Ft. Surface	Steam Temp. Deg. F.	Steam Temp. 40 Deg.	B.t.u. per Hr. per Sq. Ft. @ 2 B.t.u.	B.t.u. per hr. Bare Pipe	15 per cent. Loss B.t.u.
10	2.82	4000	11,280	344	304	608	6,858.240	1,028,736
8	2.257	2200	4,965	327	287	474	2,849,910	427,486
6	1.8	2000	3,600	312	272	544	1,958,400	293,760
Total								1,749,982*

* 1750 lb. steam.

TABLE III. AVERAGE RADIATION FROM HOT-WATER MAINS

Size In.	Sq.Ft. per Ft.	Length Ft.	Sq.Ft. Surface	Steam and Water Temp. Deg. F.	Water Temp. 40 Deg.	B.t.u. per Hr. per Sq. Ft. @ 1.8 B.t.u.	B.t.u. per Hr. Bare Pipe	15 per cent. Loss B.t.u.
18	4.71	7000	33,970	160	120	216	7,337,520	1,100,628
14	3.662	8000	29,296	160	120	216	6,327,936	949,190
12	3.333	2700	9,000	160	120	216	1,944,000	291,600
Total								2,341,418*

* 2341 lb. steam.

stant line loss, the greater line loss occurring at periods of low load on account of the increased pressures and temperatures due to lower velocities and volumes. The same is true in a less degree of the low-pressure steam system, while the loss on the hot-water system will vary from 50 per cent. greater loss in extreme weather to 50 per cent. less in moderate weather and periods of light load, that is, the loss will vary with the actual amount of heating required.

The condensation from both steam systems is of necessity wasted to the sewer while all is saved on the water system at nearly the same temperature that the steam is used in the heaters. Drain traps are generally a continual source of waste unless a force of men is maintained to keep them in proper operation. They release the water at the temperature corresponding to the steam pressure whether high or low, and the heat in the liquid is wasted. At 50-lb. pressure the water would leave the mains at about 300 deg.

In the high-pressure steam system with the great drop in pressure it would hardly be safe to omit drain traps on the mains. If the consumption of heat were low the pressure would increase, reducing the velocity, and water at a high temperature would collect due to the line loss. If a large amount of surface were turned on suddenly this water would likely cause water-hammer. It would flow along the pipe, holding its temperature until a condition of lower pressure were reached, when a portion would reevaporate. In any case the repair bills in such a system would go a long way toward balancing the loss by radiation on the hot-water system which will have the lowest repair account if properly installed at the start.

The maximum heating load will be assumed to be 75,000,000 B.t.u. per hr. and the average load will be

about 60 per cent. of this. The factor for the water system should be taken at about 55 per cent., due to the greater flexibility of the heating medium, but for ease of comparison both systems will be taken at 60 per cent. At a maximum rate of 0.3 lb. of condensation per square foot per hour on the steam system in zero weather, a connected load of 250,000 sq.ft. could be carried. The water system would require about 10 per cent. more surface at a lower rate of steam consumption to heat the same space. The average load on either system in pounds of steam per hour would be

$$75{,}000 \times 0.60 = 45{,}000 \textit{ lb. per hr.}$$

In comparing the two systems the steam system will be figured from 125 lb. gage to 50 deg. feed temperature and the cost of all water for heating will be charged. As 20 in. of vacuum will be carried on the water system in average heating weather, the feed water will be 160 deg. temperature and an initial pressure of 150 lb gage and 100 deg. superheat will be assumed as the requirements for a rate of 15 lb. per kw.-hr. at 28 in. of vacuum. It will be assumed that the steam pressure is reduced to atmospheric on the steam system and the condensation leaves the system at a temperature of 212 deg. This will not be quite true in practice as it will probably be higher. The total heat at 165 lb. absolute and 100 deg. superheat in a pound of steam is 1252 B.t.u. and the heat in the liquid above 32 deg. at 160 deg. is 128 B.t.u., leaving 1124 B.t.u. as the total amount of heat introduced per pound of steam on the water system.

For the steam-heating system the total heat at 140 lb. absolute is 1192.2 B.t.u. per lb., and the total heat in the liquid above 32 deg. at 50 deg. is 18. Thus 1174 B.t.u. per lb. of steam is introduced. Even if exhaust steam is provided from some source to heat the feed water in the case of steam heating, this heat should be charged in making a fair comparison.

The amount of heat actually introduced, including power and heating on the water system, is 50 B.t.u. less than on the steam system. The loss in the mains per pound of steam distributed under average conditions for the steam system would be

$$\frac{1{,}749{,}982}{45{,}000} = 38.9 \textit{ B.t.u. per lb.}$$

The loss on the water system would be

$$\frac{2{,}341{,}418}{45{,}000} = 52 \textit{ B.t.u. per lb.}$$

The total heat in the vapor at atmospheric pressure is 970 B.t.u., which is all that is available for heating. If a higher pressure is used by the customer the loss in the condensation will be greater and the latent heat less. As the condensation is metered the loss is entirely up to the customer. The heat in the steam system available for heating is the total heat of the steam at 125 lb. gage less that in the water at 212 deg. less the loss in the line per pound of steam, or 39 B.t.u. Thus

$$1192 - (180 + 39) = 973 \textit{ B.t.u.}$$

is available. The per cent. of heat recovered on the steam system, providing all heat less the calculated line loss is delivered to the customer, would be

$$\frac{973}{1174} = 82.9$$

In the water system the latent heat in the vapor at 160 deg. temperature and 20 in. of vacuum is 1001.6 B.t.u., and deducting the loss in mains of 52 B.t.u., the total heat delivered would be 949.6 B.t.u. per lb., giving an efficiency of

$$\frac{949.6}{1124} = 84.5 \textit{ per cent.}$$

The amount of heat available for power per pound would be the total heat at 150 lb. gage and 100 deg. superheat, or 1252 B.t.u. less the total heat at 20 in. of vacuum or 1130. This would give for power,

$$1252 - 1130 = 122 \textit{ B.t.u.}$$

and the total efficiency would be

$$\frac{949.6 + 122}{1124} = 95.3 \textit{ per cent.}$$

From these figures the water system is about 2 per cent. more efficient and in addition energy is obtained for power. This difference is due mainly to the loss in condensation to the sewer on the steam system.

The flue-gas temperature would be considerably higher in the case of the superheated steam which is at a temperature of about 466 deg., while the temperature of the saturated steam at 125 lb. pressure is 353 deg. F., a difference of 113 deg. As economizers should be installed in either case for economical operation, the total evaporation should be nearly the same per pound of fuel.

There are 5000 hr. in the heating season, as shown by the weather-bureau reports. The same evaporation per pound of coal, 8 lb., will be used for both steam and water systems. The available kilowatts from the average heating load of 45,000 lb. per hr., will be

$$\frac{45{,}000}{20} = 2250 \textit{ kw.}$$

and deducting 365 kw. for pumping will leave as net available power 1885 kw.

Under conditions of the average power load of 1885 kw., utilizing the exhaust steam under partial vacuum, the work of the condenser would not be much over 5 per cent. of the requirements of full vacuum throughout the entire heating season. The condenser is only used to regulate the vacuum on the heater due to fluctuations of both heating and power loads and this regulation can easily be accomplished while other machines are operated on the same condenser at 28 in. of vacuum. When the pumping is deducted the condensing capacity of the heating system is obtained at no expense.

If the heating and power plants were operated separately 5000 boiler horsepower would be required or 2500 for each, as the peak load of both might occur at the same time. With 75,000 lb. of steam per hr. for heating the 2250 kw. would be obtained at a steam rate of

$$\frac{75{,}000}{2250} = 33.33 \textit{ lb. per kw.-hr.}$$

which would provide for the turbines exhausting at atmospheric pressure or slight back pressure in zero weather. If 1885 kw. net was obtained from the heating steam, the boiler power for 5000 kw. peak power load would be on the byproduct plant

$$\frac{5000 - 1885}{34.5} \times 15 = 1350$$

Instead of 2500 hp.

$$2500 + 1401 = 3901 \textit{ hp.}$$

would be all that would be required for both peaks on the

byproduct plant, a saving of practically 1100 boiler horsepower in installation. As these peaks would come at the same instant, only a small percentage of the time the spare boiler power would be more than ample under average conditions.

The cost per horsepower installed for the high-pressure steam system will be as follows:

Boilers	$17
Building	10
Piping	8
Economizer and draft equipment	10
Total	$45

The cost of water at $1 per 1000 cu.ft. on an average load of 45,000 lb. per hr. will be

$$\frac{45{,}000 \times 5000}{62.5 \times 1000} = \$3600 \textit{ per season}$$

The cost of 10c. per 1000 lb. of steam at 8 lb. evaporation given by Mr. Ballard is low for most localities and represents a coal cost per short ton of $1.60. The total investment for the steam system will be:

2500 boiler horsepower at $45	$112,500
Cost of mains as given	65,550
Total	$178,050

Assuming a byproduct plant with an average load of 2000 kw. and 5000 kw. as the maximum, or a load factor of 40 per cent. as per Mr. Ballard's paper, the cost of the hot-water heating equipment will be

Pipe	$52,910
Fittings, valves and expansion joints	17,303
Excavation and back fill	13,000
Miscellaneous material	12,400
Repairing 2000 ft. at $1.20; 13,700 ft. at $1.15	18,155
Conduit and covering	48,566
Steam fitting labor	6,000
Heaters, pumps, piping in power house	30,000
	$198,334
1500-hp. boilers at $45	67,500
	$265,834

In utilizing exhaust steam from power the fuel charge for heating will be the difference in steam required to operate the engines on reduced vacuum at an average in this case of 20 lb. per kw.-hr. and that required to operate the engine for the same power load at full vacuum, or 15 lb. The operation at full vacuum represents the exhaust steam saved for heating. The full rate is charged for the pumping as this is practically live-steam operation.

The cost of operation on the steam system would be

Interest and depreciation, 10 per cent	$17,805
Extra maintenance on steam system due to steam traps	1,000
45,000 lb. steam 5000 hr. at 10c.	22,500
Water	3,600
Plant labor—three shifts, three firemen at $60 per month for 7 months	5,000
One man in charge all the year at $1200	1,200
Total	$49,905

The cost of operation on the hot-water system would be

Interest and depreciation 10 per cent	$26,583
Cost of pumpage 365 kw.-hr. at 20 lb.—10c. per 1000 lb. for 5000 hr.	3,650
1885 kw.-hr. at (20—15 lb.) 5000 hr. at 10c. per 1000 lb. of steam	4,712
Labor for heating plant—will be three shifts, two firemen for 7 months at $60 per month	2,520
Total	$37,465

This shows a saving in operating cost for the hot-water byproduct over the steam system of

$$\$49{,}905 - \$37{,}455 = \$12{,}440$$

per year, or a saving of 25 per cent. On the difference in investment the $12,440 would pay

$$\frac{12{,}440}{265{,}834 - 178{,}050} = 14.17 \textit{ per cent.}$$

If the price of coal should increase 50 per cent., or to $2.40 per ton, the additional cost of fuel on the steam system would be about $11,250 per year. The total operating expense would then be $61,155 On the water system the additional charge would be $4200 per year, making the total $41,665. Then

$$\$61{,}155 - \$41{,}665 = \$19{,}490$$

$$\frac{19{,}490}{87{,}784} = 22.2 \textit{ per cent.}$$

the saving on the additional investment.

It is well within the realms of possibility that the price of steam coals will increase in cost in a greater ratio than domestic sizes and the analysis shows that this increase in cost of fuel will not detract from the financial standing of the hot-water byproduct plant, while it might reduce the high-pressure steam plant to a nonpaying basis.

If 250,000 sq.ft. is the total connected load on the hot-water and steam systems the cost per square foot per season with coal at $1.60 and $2.40 per ton of 2000 lb. without any profit or administration charge are found in Table 4. The cost per 1000 lb. of steam distributed with an average load of 45,000 lb. of steam per hour is also given for both systems. Taking 7 lb. evaporation as the equivalent efficiency of domestic coal, Table 4 gives the price per ton that the latter would cost to equal the central-station cost without profit or administration charges.

TABLE IV. COSTS WITHOUT PROFIT OR ADMINISTRATION EXPENSES

	Coal $1.60 per ton 2000 lb.			Coal $2.40 per ton 2000 lb.		
	Cost per 1000 Lb. Steam, Cents	Price Domestic Coal per Ton Equivalent to Central Station Cost, Dollars	Cost per Sq.Ft. with 250,000 Sq.Ft. Connected Load, Cts.	Cost per 1000 Lb. Steam, Cents	Price Domestic Coal per Ton Equivalent to Central Station Cost, Dollars	Cost per Sq.Ft. with 250,000 Sq.Ft. Connected Load, Cts.
Steam System	22.18	3.10	19.96	27.2	3.81	24.46
Hot Water System	16.65	2.33	14.986	18.52	2.594	16.7

A successful central-heating plant should be able to distribute the heat to the consumer at a cost including profit and all charges at about the price of domestic coal. In giving the price per square foot per season for the hot-water system it is understood that the steam surface being at a slightly higher temperature than the water will do more work, but as an increase in the connected load would decrease the price per square foot, both water and steam were taken at the same figure. The byproduct plant appears to be the best investment. An increase in steam consumption on the turbine at full vacuum would not destroy the relation as the saving would be greater on the same percentages.

No live-steam operation is charged on the water system and none would be necessary if the power load was as stated. If 50 per cent. of the steam were required direct from the boilers 100 days of 4 hr., or 22,000 lb. for 400 hr. at 10c. per 1000 lb., it would amount to only $880, which would not seriously affect the final conclusions.

❖

In honor of the recent visit made to Germany by the American Society of Mechanical Engineers upon invitation of the Verein Deutscher Ingenieure, the latter organization has prepared a guide book of 171 pages. This book, which is printed in English and German on facing pages, describes the points of interest in each locality to be visited, laying particular stress, of course, on matters relating to engineering and technical education. The itinerary shows that the American delegates arrived at Hamburg on June 19 and that the official visits and receptions terminated at Munich on July 3.

EDITORIALS

N. A. S. E. Convention

The old question, which was presumably settled several years ago, regarding the frequency of the national conventions of the National Association of Stationary Engineers, has been resurrected. Shall they continue to be held annually or only every other year?

At the recent state meeting in Milwaukee, as noted on another page, National Secretary Fred Raven, mentioned the principal argument for biennial conventions, the money saved in railroad fares for the delegates being available for other useful purposes, such as educational work, publicity in relation to the need of boiler inspection and engineers' license laws, etc.

We do not quote the secretary with the intent of giving the impression that he has declared himself in favor of the biennial as against the annual meeting. We do not know his position on the question. We are not sure of our own, but if discussion is in order on the topic we have a few thoughts we would like to submit for consideration.

As Mr. Raven also said the association needs to be uptodate and employ modern methods quite as much as any business institution if it is to progress. On that point certainly all are agreed. Now then, can it be shown that it is better business to adhere to the annual-meeting idea?

Take as an illustration a somewhat parallel case, one related to the advertising policy of a growing business. Here is a man making an excellent product only fairly well known, its worth not being appreciated by a very large share of the possible number of users. Because the orders do not happen to be coming in as briskly as he would like, suppose this man says he will curtail his advertising and save money.

What happens? Well, what happens when any bad conditions are neglected? "The operation was a success—but the patient died." When your steam pressure is falling, you do not cease firing to save money. If you want more steam you must use more coal. If the manufacturer wants more business he must extend knowledge of his product. If the N. A. S. E. wants to continue to grow in size and influence it must keep itself before the public, and is not the annual meeting one very good way of doing this? True, it costs money, but is not this money well invested? If meetings were held less frequently, would not the actual receipts of the association dwindle by more than the difference saved?

To say nothing of the other benefits of the conventions, their educational value, extension of the individual's circle of acquaintance among those identified with the same profession, promotion of good fellowship, discussion of questions of scientific interest or relating to general welfare of the profession, as by legislation, etc., putting it purely on a selfish, or at any rate commercial basis, do not the conventions attract new members as much or more than any other one thing connected with the association?

No, this line of talk is not forgetting the state conventions that would go on just the same, but there is a far greater enthusiasm brought out by an annual meeting than can be at the state meetings. There is a feeling of pride and gratification that comes into the heart of a man when he gazes about him and sees the hundreds and hundreds of men from all over the country all enlisted in the same contest of harnessing the forces of nature to the will and need of mankind. Rather high-sounding words, but really they do not seem so when one gets to the convention and gets full of that Fourth of July feeling.

As we said earlier in this editorial, we do not wish to seem to be advising one course or the other in the matter. All we are concerned for is the best interest of the association in whatever way it may be served. We are confident that the association's officers will do the wise thing whatever that proves to be. At the present time of writing we are from the extreme interior of Missouri, however, on the question, but we are entirely open to conviction.

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Purchasing Coal

In discussion following a paper on boiler-furnace operation, read before the Association of Building Owners & Managers, at its recent Cincinnati convention, the question: "Is it advisable, when buying coal in such amounts as three thousand tons per year, to purchase it on the British-thermal-unit basis?" was asked. Of course, the reader of the paper said it was. Let us see why.

At four dollars a ton (2240 pounds), and using three thousand tons per year, the heat loss due to the difference between coal having an average of 13,500 thermal units per pound and that having an average of 11,000 is 11,760,000,000 thermal units. Considering that seventy per cent. of the heat units in the coal is represented by the water evaporated; seventy per cent. being a fair average under ordinary good operating conditions, the money loss is over fifteen hundred dollars; which is much more than would be necessary to defray the expense of analysis.

The big point about this question is that it is another indication of the fact that many engineers and managers do not make any effort to determine, even approximately, what kind of coal they are buying. So many seem to think because they buy less than several thousand tons of coal per year, that it would not pay to have the fuel analyzed. And because they do not feel justified in doing so, they take no steps whatever to keep track of the quality of coal received, except to occasionally consider the complaints of the firemen after they have repeatedly reported that it is almost impossible to keep up steam with the grade they are using.

One may get the average coal dealer to promise many things when he is after a big order, but unless he is held by something stronger than a mere promise, the grade of coal furnished is likely soon to fall below that agreed upon and furnished in the beginning. How may one know how inferior the coal is unless provided with means to find out? The firemen will not complain until their

work becomes well-nigh intolerable under conditions which they know should not require such hard work if the normal grade of coal were being used.

There are few plants burning so little coal as not to warrant a wheelbarrow and a couple of platform scales for weighing the coal and feed water for making evaporative tests.

It is not necessary to make such evaporative tests elaborate to find out if there is a material difference in the grade of coal furnished. One need simply try to have the conditions the same for one test as for others; differences in the quality of the coal will be readily observed.

The engineers of a community may find it difficult to get the coal dealers to acquiesce to supplying coal on specification, but when the dealers find that those who buy the bulk of their product, are determined to buy it on something more than a "hit-and-miss" basis, they will meet the new conditions.

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New York Smoke Ordinance Unconstitutional

On Friday, July 18, the Court of Special Sessions declared "unconstitutional and unreasonable" Section 181 of the Sanitary and Health Code under which the Health Department of New York City had filed nearly one hundred complaints against the New York Edison Co.

This section reads: "No person shall allow dense smoke to be discharged from any building, vessel, stationary or locomotive engine, or motor vehicle, on any place or premises within the City of New York or upon the water adjacent thereto within the jurisdiction of the city."

The decision reads in part: "The section is unreasonable and arbitrary, because of its unqualified and sweeping character, condemning as a nuisance a thing that may or may not be a nuisance, and because it makes no provision for cases where compliance is impossible."

"The use of furnaces and boilers and the consequent emission of smoke is doubtless necessary to the enjoyment of the ownership of practically every building within the city of New York." As a further reason for calling the section unreasonable the decision states that "it applies generally to all emission of dense smoke and contains no qualifying clause exempting from its operation cases where it might possibly be shown that no known fuel consumer or device could have prevented such discharge."

This latest case again brings out the fact that the smoke ordinance of New York City is little better than no ordinance at all. With all due respect to the court judges they cannot be expected to know as much about the combustion of fuels as do combustion experts, and the interpretation of a smoke ordinance should be in the hands of those competent to weigh the enormity of a violation against the offender's ability to comply.

The press of New York City bewails the decision of the court and urges Commissioner Lederle to appeal the case. The decision may be an ill wind that will blow the city much good.

Every now and then there is a many-voiced protest that this or that company is violating the smoke ordinance. This will continue so long as there is no organized smoke-inspection department to work along lines similar to the one in Chicago.

Commissioner Lederle undoubtedly feels that as long as he is health commissioner he will endeavor to prevent under the present ordinance the presence of smoke which is injurious to health. He also, perhaps, feels that if a smoke-inspection department were created it would not come under the direction of the health department, and for that reason he is not likely to encourage the creation of that much-needed department.

Now that the ordinance has been declared unconstitutional, thereby putting an end to the prosecution of offenders for violation, the Bureau of Municipal Research should bestir itself and investigate the smoke situation and report the necessity for a smoke-inspection department. Otherwise, Commissioner Lederle will appeal the case and, undoubtedly, a higher court will order an amendment of the ordinance to specifically cover points that it now only generally covers.

An amended ordinance will not better conditions so long as there is no recognized department and trained inspectors to coöperate with offenders and assist them in obtaining the smokeless combustion of the lower grades of fuels that must be used under boilers to obtain maximum economy of operation.

Nearly every city of any importance is now taking practical steps to abate the smoke evil. Why should the metropolis of the country, with its millions of population, its beautiful buildings, its many large power houses, employing excellent engineers, and its location which makes easy the procuring of any kind of coal, continue to hold sacred an archaic smoke ordinance?

New York City has the money, the means and the men to organize a smoke-inspection department that will meet any condition of smoke nuisance. It is the social and moral duty of the Bureau of Municipal Research to show the city officials how to organize such a department.

❖

Two practical articles dealing with the attainment of the highest economy in the operation of steam engines, are to be found in this issue. The first, page 157, declares that valve leakage more than increased cylinder condensation accounts for falling off in economy at light loads. This article is of more concern to designers than operators. Both should be interested in the experiments described on page 159 to determine the effect on economy of varying the compression in steam engines. The results controvert earlier beliefs in that they show that compression up to about 45 per cent. of the initial absolute steam pressure is advantageous. Those who have not already read these two articles will do well to do so, for they are instructive.

❖

"The thing I like most about my chief engineer," said the general manager of a large plant, "is that when things go wrong he reports them to me, but not until he has remedies for, or has corrected the troubles."

That's the way with all good managers. No one likes a subordinate that waves a red flag in announcement of trouble. It's not so much a question of what's wrong as how soon will things be running again.

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Can anyone tell why in most plants there is one engine or pump, built like the rest, but always giving trouble as though it were hoodooed?

READERS WITH SOMETHING TO SAY

Cutting Pipe under Difficulties

Two pipe bends were ordered for connections from two new horizontal return-tubular boilers to the main steam header. Between the header and the pipe bend a 6-in. valve was to be placed, but through an oversight in ordering the piping the space to be occupied by the valve was not taken into consideration; consequently the bends ordered were too long. The mistake was not discovered until the pipe arrived at the plant. In checking up the piping it was found that the bends were too long by just the length of the valve.

Being anxious to get the new boilers in operation, we decided to get out of the trouble if possible without returning the bends to the makers.

Not being able to swing the bend with the pipe machine in its regular position, we were obliged to stand the machine on end so that the bend would have plenty of room to turn. The machine was held in position by being firmly fastened to the floor. The change of position of the pipe machine made it necessary to change the drive which was easily accomplished. The pipe was cut and threaded in much quicker time and at less cost than if returned to the makers.

ROBERT LOVE.

Baltic, Conn.

Piping for Feedwater Heater

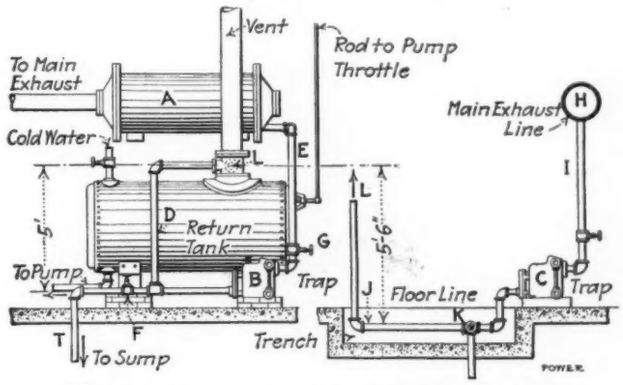
The illustration represents the heater and return-tank layout of a new plant which is supposed to be in accordance with the latest ideas in power-plant design. The point that I wish to bring out in the installation is the method of draining the feed-water heater.

This heater is connected up somewhat different than most closed heaters. We are all familiar with the closed heater *A* placed in the exhaust line from the engine. After the steam has passed through the heater it goes to the heating system or the atmosphere. In this plant the exhaust is used for heating during the cold weather, the returns being piped to a vacuum pump which discharges into the return tank. The pipe supplying the heater with steam is taken off at one side and the condensation is supposed to be taken care of by the small pipe which is shown leading to the trap *B*. From the trap it is supposed to flow up the pipe *D* into the return tank and rise to a point 5 ft. above the trap before it is discharged. By opening the valve *F* this water can be discharged to the sump.

It is desirable to save all the condensate as the cylinder oil has been separated from it and in this plant the water has to be purchased from the city. Since the vacuum system of heating is used, there are times when the pressure on the main exhaust line and also in the heater is but slightly higher than that of the atmosphere. Also the heater is located about 115 ft. from the engine so that there must not be much resistance to the flow of the steam, if any, if it is going to raise the temperature of the feed water to 212 deg. This has been well provided for by an exhaust main of ample size, but the pipe *E* leading from the heater to the trap *B* is only 1½ in., which is large enough if there is sufficient pressure to force the steam through.

Supposing that there is force enough to make the water flow to the trap *B* it is going to require considerable more pressure to force it up the pipe *D* and then over into the return tank. Thus if there were but ½ lb. pressure on the exhaust line there will be a poor circulation in the heater, and this has worked out in practice in just this way, for, although the heater is higher than the return tank, there is enough friction added to the weight of the column of water in the pipe *D* to keep the steam from discharging to the trap *B* unless there is a pressure of at least 2 lb. on the heating system.

When no steam is required for heating, in order to use



HEATER CONNECTIONS IN MODERN PLANT

the trap at all it was necessary to keep the back-pressure valve closed and maintain a pressure of 2 lb. on the engine just to heat the feed water; so the valve *F* was opened and it was found that enough steam would flow through the heater to heat the water at the pressure of the atmosphere, but this caused an offensive amount of vapor to rise into the pump room from the sump well so the piping was changed to better the conditions in this manner.

The riser *D* was removed and the horizontal part of the pipe connected to the pipe *E* at a tee so that when no steam was needed for heating, it could blow through into the heater and what is condensed will be added to the feed while the vapor will be carried to the roof vent pipe. When this is operating in this way the valve *G* is kept closed, but when it is desired to run the heating system, the valve *G* is opened and a valve placed in the connecting pipe is closed and the pipe *F* leading from the trap *B* is continued to the main return line of the system so that the trap will discharge into a vacuum and the water cannot back up, thus keeping the heater always full of steam.

The arrangement at the right was expected to drain the main exhaust line by the pipe *I* which leads into the

trap *C* and formerly discharged into the return tank through the pipe *J*, being connected to the vent pipe at the point *L*. Under very low pressure the trap *C* was always full of water and did not do its work properly so the pipe *J* was connected to the main return pipe of the heating system and thus the discharge from the trap *C* is taken care of and when there is no steam required for heating, the water is allowed to run to the sump through a valve placed at the point *K*.

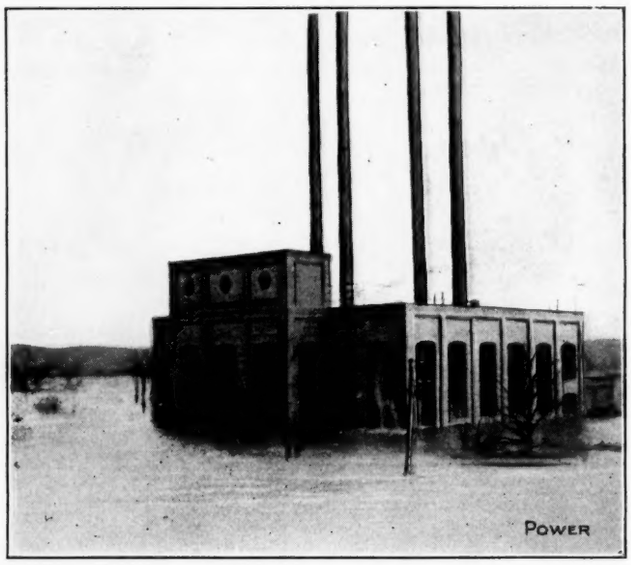
G. H. KIMBALL.

East Dedham, Mass.

Operating Plant during Flood

The photograph shows the plant of the Electric Light & Ry. Co., of this city, taken at the crest of the flood. At the time the picture was taken, the water was 13 ft. above the engine-room floor, and about 12 ft. above the boiler-room floor.

The water was kept out by bulkheads built in the door



CRIBS AT DOORWAYS KEPT WATER OUT OF THE PLANT

openings. Six pumps were in constant operation, pumping out the water that leaked into the building.

The power plant was in constant operation throughout the flood, which speaks well for the entire operating force, which never left the building during the ten days of the flood.

FRANK G. BROWN.

Portsmouth, Ohio.

Orsat or CO_2 Recorder Best?

The fact being established that flue-gas analysis is inductive to higher boiler efficiency, progressive engineers are now analyzing flue gases, the majority using either the CO_2 recorder or the Orsat apparatus. There appears, however, to be a difference of opinion as to which of these two is the better. While there remains no doubt but that a recording instrument of any kind is usually preferable to a nonrecorder, the CO_2 recorder as it is today leaves much to be desired. There is only the recording feature to recommend it. While a practically continuous record of CO_2 is desirable, CO_2 alone does not indicate efficiency.

The analyses should show a practical absence of CO as well as a large percentage of CO_2 to indicate high efficiency. As the recorder does not show the CO, which is detrimental to efficiency, using this instrument would, it seems to me, be like taking indicator cards, and omitting the atmospheric line.

The Orsat apparatus, on the other hand, has several features to recommend it. While it does not give a continuous record it can be used to analyze a reliable average sample. If the boilers were operated carelessly, say, several hours during the watch, it would, of course, be almost impossible to tell just when this carelessness occurred, but at the end of the watch the average would naturally be lower than it should be. A sampling bottle properly set up would be more difficult to cheat than a recorder.

With the Orsat a more complete analysis is possible, for it gives the CO_2, the O and the CO. There are other disadvantages about the recorder. One is the fact that the continuous flow of gas through the solution tends to weaken this solution, the result is that after the recorder has been in operation a number of hours, the solution is not strong enough to absorb the entire amount of CO_2 in the gases, and the percentage of CO_2 will vary, though the conditions in the furnace remain practically the same.

There is also the first cost of the recorder to be considered along with the fact that it is a delicate piece of apparatus easily broken when an inexperienced hand gets at it. The Orsat apparatus, on the other hand, costs but little, and one apparatus is all that is necessary, irrespective of the number of boilers. Sampling bottles are cheaply replaced if broken.

A. POHLMAN.

Brooklyn, N. Y.

Testing for Approximate Heat Value of Coal

The question sometimes arises as to what will be the cheapest coal to use in a given plant. What coal will evaporate the most water per dollar under actual running conditions?

To answer the above question I have frequently made the following simple test, which may be made in almost any plant without much expense for equipment. This test is of value only as a comparison test as there is an error in measuring the feed water, but this error can be made practically constant for the different tests and its only effect will be to show a greater evaporation than actually takes place, and for this reason I shall call the evaporation as found by this test the apparent evaporation.

The equipment necessary for the test includes scales to weigh coal and ashes, a thermometer placed in the feed-water heater or feed line and a counter for recording the number of strokes of the pump. Before starting the test, see that the pump is properly packed, that the valves are in good condition, that the pump is also in ordinary working order, and that the blowoff valves are tight. The test should cover a period of at least 24 hr.

When starting the test have the ashpits clean and note the condition of the fire and the height of the water in the glass. Note the pump revolution counter, thermometer and steam-gage readings. Repeat these readings

every hour during the test. Weigh the coal used and all ash and clinker formed during the test period. At the end of the test have the ashpits clean and the height of the fire and water the same as when the test began.

The reading of the revolution counter at the beginning of the test subtracted from the final reading multiplied by the total pump displacement in cubic inches for one indication of the counter, divided by 231 and multiplied by 8⅓ gives the pounds of water apparently evaporated. The average steam pressure and feed-water temperature are next found by adding together the hourly readings and dividing by their number. From a table of factors of evaporation the factor for the average steam pressure and feed-water temperature is found and the pounds of water apparently evaporated multiplied by this factor gives the pounds of water apparently evaporated from and at 212 deg. F., which divided by the total pounds of coal used gives the pounds of water apparently evaporated per pound of coal. This result multiplied by the number of pounds of coal that one dollar will buy gives a standard of comparison.

The percentage of ash and clinker is easily found by dividing the weight of ash and clinker by the weight of the coal.

MYRON D. PLACE.

Foxboro, Mass.

The Value of Graphite

Recently I had impressed on me the good qualities of graphite for cooling and preventing the cutting of engine cylinders and bearings.

The high-pressure cylinder of a horizontal triple-expansion tandem engine gave trouble with the lubrication of both Corliss valves and cylinders when the full superheat of 450 deg. F. with 160 lb. pressure was supplied to the engine which was considerably overloaded. This engine was built about 17 years ago and is not of a favorable design to deal with superheated steam.

It was decided to try mixing a little graphite with the cylinder oil in the proportion of one ounce to the gallon of oil with the results that after nine weeks' application the valves now close quicker and the surfaces of all rubbing parts are very smooth and we now run with full superheat and no signs of the valves or cylinder walls cutting or sticking.

The bearing for an 18x13-in. crankshaft had been adjusted and ran cool for the first hour, after which the "fun" suddenly started. Castor oil, which is used on these bearings, simply ran out like water, or, at least, that ran out which did not go up in vapor. It was seen through the hole in the top of bearing cap that the bronze and shaft were cutting badly, so as a last resource graphite was mixed with cylinder oil in the proportion of a small box (2x2x¾ in.) full to a pint of oil. On pouring in the first dose the results were very pleasing. The bronze immediately stopped tearing and the shaft commenced to polish. This treatment was continued for nine hours, the graphite being cut down to the above quantity added to a gallon of oil after the first two hours, the cylinder oil being gradually cut out after about four hours and castor brought into use. During this time a stream of cold water was kept running on the crank webb and another on the shaft to draw the heat out. No water being put into the bearing. Now this bearing has a better surface than it ever had and is running cooler than before.

The nature of our output requires that the engine be kept running at full speed all the time. We estimated that about $750 was saved by not stopping the engine to cool the bearing, which would have been necessary only for the graphite.

R. EDWARD.

Rochdale, Eng.

Air-Compressor Trouble

Herewith are diagrams taken from the low-pressure side of a two-stage compressor, with cylinders 18¼x11¼x14 in., compressing air to a final pressure of 100 lb. gage. Lately the intercooler pressure has fallen considerably and varies with the speed, as shown by the diagrams. It is evident that there is a leak on the low-pressure side of the intercooler and I have tried the indicator to see whether or not it would point to the trouble. Before using it the gage was tested on a dead-weight tester and its accuracy assured. Both diagrams were taken from the same cylinder with the same reducing motion, one at

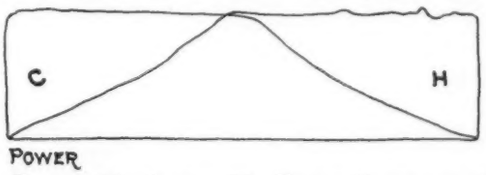


FIG. 1. R.P.M., 25; SCALE, 20; FINAL PRESSURE, 99 LB.; INTERCOOLER PRESSURE, 12½ LB.

25 r.p.m., and the other at 145 r.p.m. Each diagram fairly represents the extreme condition, as the highest speed of the compressor is about 156 r.p.m. The difference in the lengths of the diagrams is $2\frac{11}{16} - 2\frac{1}{2} = \frac{3}{16}$ in. After taking the diagrams I found that this was due to the cord, which proved to be more or less elastic.

The compressor is running under good conditions, so far as cooling water for jackets and the intercooler is concerned, and the air, when the compressor is running slowly, leaves the intercooler at a lower temperature than the outside atmosphere at this time of the year, and does not rise much, if any, above it when running up to full speed.

The compressor is an old machine, although it is in fairly good condition. The present trouble appears to be

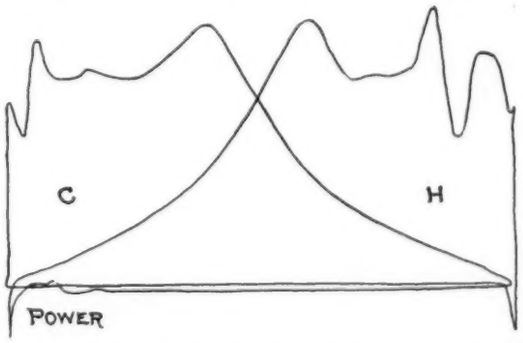


FIG. 2. R.P.M., 145; SCALE, 20; FINAL PRESSURE, 92 LB.; INTERCOOLER PRESSURE, 20

a leak either through the packing or valves. I would be glad to hear from readers on the trouble.

W. L. SHAWKEY.

Youngstown, Ohio.

ENGINEERS' STUDY COURSE

Heat Lost in Flue Gases—II

Estimating Nitrogen

The flue-gas analysis as made according to the directions given in previous lessons gives the percentage by volume of carbon dioxide (CO_2), oxygen (O) and carbon monoxide (CO). In addition to these there exists water, in the form of superheated steam, nitrogen and sometimes hydrogen and various hydrocarbons. The steam condenses and only a little of it gets to the flue-gas apparatus. All the other constituents except the nitrogen form a small percentage of the total volume. Partly because of this fact and partly because it is difficult to detect and measure them, they are usually ignored entirely. Thus, the difference between 100 and the sum of the percentages of CO_2, O and CO is taken as the percentage of nitrogen (N) contained in the flue gas.

To illustrate, if the flue-gas analysis shows 11 per cent. CO_2, 8 per cent. O and 0.8 per cent. CO, the percentage of nitrogen in the flue gas would be taken as

$$100 - (11 + 8 + 0.8) = 80.2$$

This practice is followed because there is no liquid or chemical solution capable of combining and absorbing nitrogen like CO_2, O and CO are combined and absorbed.

With equal temperature and pressure CO_2 has the same volume as the oxygen required for its combustion. That is, if it requires 30 cu.ft. of pure oxygen at a certain temperature and atmospheric pressure to completely burn a pound of pure carbon the CO_2 formed will have a volume of just 30 cu.ft. when cooled down to the temperature at which the oxygen was measured. The volume of carbon monoxide is twice that of the oxygen required in its formation. Thus, the sum of the percentage of CO_2 and O and one-half the CO should always equal 21 (the percentage by volume of oxygen in the air) when carbon alone is the combustible. But in the average plant carbon is rarely the only combustible in the fuel; it is almost always accompanied by hydrogen.

When hydrogen burns it forms steam which practically all condenses and never gets to the flue-gas apparatus. Thus, the hydrogen sidetracks the oxygen of the air required for its combustion but leaves the nitrogen to pass on to the flue-gas apparatus along with the nitrogen of the air required for the combustion of the carbon. The result is that the sum of the three oxygen constituents of the flue gas (CO_2, O, CO) do not add up to 21, but to some smaller figure, the exact value of which depends, of course, upon the quantity of available hydrogen contained in the fuel.

Using the data given in the last lesson for coal and flue-gas analysis in this lesson, the formula for estimating the weight of flue gases formed per pound of fuel burned works out thus

$$W = 3.032 \times 0.779 \left(\frac{80.2}{11 + 0.8}\right) + (1 - 0.063) = 17 \text{ lb.}$$

Assuming that the temperature of the gases as they leave the boiler is 520 deg. F. and that the temperature of the air entering the furnace is 70 deg., the formula for calculating the heat lost up the chimney in the flue gases would work out thus

$$L = 0.24 \times 17\ (520 - 70) = 1836\ B.t.u.$$

If the coal in the foregoing illustration had a heat value of, say, 13,200 B.t.u. per pound, as fired, the percentage of heat lost with the flue gases would be

$$\frac{1836}{13{,}200} \times 100 = 13.9\ \textit{per cent.}$$

Ratio of Air Supplied to Air Required

It is often desirable to know what the ratio is between the amount of air actually supplied per pound of coal and the amount theoretically required. This can be estimated by means of the following simple equation:

$$R = \frac{N}{N - (3.78 \times O)}$$

where R equals the ratio of the air supplied per pound of fuel to the amount theoretically required and N and O equal the percentage by volume of the nitrogen and oxygen, respectively, in the flue gases, as shown by the analysis.

Using the data previously given the ratio would be

$$R = \frac{80.2}{80.2 - (3.78 \times 8)} = 1.6$$

This means that for every pound of air required by the coal 1.6 lb. was supplied. Hence, the percentage of excess air is

$$\frac{1.6 - 1}{1} \times 100 = 60$$

Measuring Flue-Gas Temperature

Thermometers with a range up to 212 or a little higher are cheap enough that no difficulty need stand in the way of measuring the temperature of the air entering the furnace; that is, the air in the boiler room. But as the temperatures to be measured get beyond 500 deg. F., which is often the case with flue gases, the cost of the pyrometer (the name used for high-range thermometers) begins to be an important consideration. In some plants the expense of purchasing a standard form of pyrometer may not be warranted, in others it may be impossible to show the management the advantage of keeping constant records on the flue-gas temperature. In such cases the engineer may and should for his own benefit provide himself with the means of making an occasional observation of the flue-gas temperature. This he can do at reasonably moderate cost.

Home-Made Apparatus

Numerous means are available but only one will be presented here. The required equipment consists of the bomb shown in Fig. 2, and a chemical thermometer having a range of 1000 deg. F. The latter may be procured of any chemists' glassware-supply house at a cost of from $5 to $10.

The bomb consists of a length of 4-in. pipe capped at one end and a length of ¼-in. pipe extending through the center, as shown, to serve as the thermometer well. Sand is packed in about the ¼-in. pipe and held in place by means of a layer of plastic asbestos at the open end. The length of the pipes is determined by the length of the thermometer. The ¼-in. pipe should be of such a length that when the thermometer is inserted it will be inclosed as high as the 400-deg point on the scale. The 4-in. pipe should be 6 in. longer, as shown. A bail of wire or thin, narrow flat iron attached as shown completes the bomb.

When it is desired to take the temperature of the flue

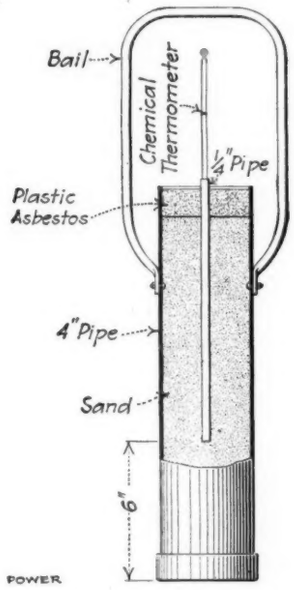


FIG. 2. SAND BOMB AND THERMOMETER FOR MEASURING FLUE-GAS TEMPERATURE

gases, the bomb (without the thermometer) is hung in the middle of the path of the flue gases as near to the point at which they leave the boiler as conveniently possible. After a half to three-quarters of an hour the bomb is removed and the thermometer inserted as quickly as possible. In a few seconds the thermometer will indicate the temperature of the sand which will be practically the same as that of the flue gases.

PRACTICE PROBLEMS

From the following data, estimate the percentage of excess air supplied to the furnace, the number of B.t.u. carried up the chimney by the flue gases and the percentage of loss based on the heat value of the coal.

Proximate analysis of coal, moisture, 4.5 per cent.; volatile, 18.6 per cent.; fixed carbon, 65.2 per cent.; ash, 11.7 per cent.; B.t.u., 13,160. Flue-gas analysis, CO_2, 6.3 per cent.; O, 11.7 per cent.; CO, 0.4 per cent. Temperature of air in boiler room, 75 deg.; temperature of gases leaving boiler, 515 deg.

If the above coal costs $3.10 per net ton (2000 lb.) and if under the present conditions 6200 tons are consumed per year what would be the yearly saving effected if the average percentage of CO_2 were raised to 10.7 and the O reduced to 7.3, other conditions remaining the same?

OVER THE SPILLWAY

JUST JESTS, JABS, JOSHES AND JUMBLES

Uncle Sam has built a brand new government dredge to be used in the Jacksonville, Fla., district, and, says a local paper, it is equipped with "two Heine compound condensing boilers." Every time we read anything like this we are reminded of William Tell's brother Wat.

⁂

In its "Instructions to Foremen," a mining company says: "If you see a man violating your instructions or taking chances that make him liable to injury, deal with him so as to make it certain he will not again disobey orders." Might order him shot at sunrise or drawn and quartered. If these are irremedial, why not fire him?

⁂

A writer in the "Electrical World," describing the Woolworth Building elevators, says he tested the safety buffers with the car descending at the rate of 1000 ft. per min. and "found it perfectly safe, and in fact, not unpleasant."

Elsewhere he states that if one of the high-lift elevators were to drop from the top of the tower, it would attain a speed near the bottom of the shaft of nearly 140 miles per hour, equivalent to over 12,000 ft. per min. Wonder if the sensation then might be "not unpleasant."

⁂

"Punk, piddy plunk—piddy punk, plunk, plunk,
O, the tail of the spineless pine!"

Something like this crawled in our ears while down to a seaside resortery t'other day and started racing over our tympanums at 1109 r.p.m. Before the end of the 24-hour run, we'll be gormed if we didn't wish the pianner, its operating engineer, boiler, flywheel and cylinder head would go up in the air to once. (At the time, we consigned the whole nerve-racking business to another direction —down.) If you're going to take a day off to rest, first find out if the ragtime fiend holds forth in that place; then go somewhere else.

⁂

It's a poor time to borrow money, but a fine time to lend it, says a contemporary. Yes, and it's warmer in summer than it is in winter, and be good and you'll be happy, and— Oh, shucks! It's all right to keep hammering home the truth, but this rubbing-it-in business sure scratches the surface of our good humor. Get something new, or run on your spare units now and then.

⁂

"Some shop grease, with some pay and prospects, is preferable to an unspecked hand that is certain to be empty 48 hours after every payday," says the "Saturday Evening Post." This paper further opines that the clean-collar superstition is slowly subsiding. It looks like the lay press was beginning to understand us. Not long ago, if an able, well-to-do man donned overalls and sported a smudge on his nose, he was either posing or just trying out his one last opportunity to forget that he had more money than he could spend. Now, some of our very best young fellows are toting tin dinner-pails and smearing their aristocratic forms with real dirt. And what shocks many is that the young fellows are enjoying the plunge, are becoming manly men, and are on the right road to learning something useful.

⁂

Goodby to the tungsten lamp if this comes to pass. A West Virginia inventor has a secret chemical bath for the common and garden variety of housefly whereby is produced a light which surpasses the tungsten incandescent. We are somewhat in the dark as to its substitute when this fly has lit out during the long nights of the drear and dread winter —but why anticipate trouble?

⁂

All that has been said about the recent Stamford railroad wreck seems to have resolved itself into the fact that the New Haven road is most culpable in that it allowed an engineer of limited experience to run a fast train unaccompanied. Let us hope that Engineer Doherty will yet be given a fair chance to prove his worth and gain that experience which he undoubtedly has been striving for. Save that kick until he is on his feet.

Heating Engineers Meet at Buffalo

Hotel Statler was headquarters for the nineteenth semi-annual meeting of the American Society of Heating and Ventilating Engineers, which was held in Buffalo on July 17, 18 and 19.

On Thursday morning President John F. Hale called to order the first session and delivered a short address of welcome. The revised laws under which the society is now operating calls for 12 directors instead of 10, so that more work can be done, and he made an earnest appeal to all members for a more hearty coöperation in the association affairs. Routine business, the reports of the Illinois and New York chapters and those of the various committees occupied the entire morning session.

PROFESSIONAL SESSIONS

The first paper, "Time Analysis in Starting Heating Apparatus," was read by Ralph C. Taggart at the opening of the afternoon session. The author showed how to figure the time that is required to warm up heating apparatus. Proper allowance must first be made for the starting of the fire or bringing it up to its normal operating condition. The heat in British thermal units which is given to the iron and water in the boiler before steam is produced was first considered, and then the time required to heat the piping and the radiation in the heating apparatus itself.

Formulas giving the time required in both steam and hot-water systems were developed by the aid of calculus and, to illustrate the use of the equations, certain examples and curves were worked out in the paper.

H. M. Hart followed with a paper on the "Expense of Operating Heating and Ventilating Plants." Methods for computing the cost of operation in advance of construction were given for four different classes of building, namely, residences, apartment buildings, school buildings and office buildings. The data on residences and apartments was conflicting and unsatisfactory. Two school buildings were compared and also two office buildings of about the same size and type, one having its own power plant and the other operating a heating plant only. The former showed a saving of $26,801 over the latter.

In a short discussion on "Ventilation and the Open Window," M. S. Cooley gave some results of tests conducted for the Government at Washington. An interesting paper on "Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts," was then read by Frank L. Busey. The author gave the results of two sets of experiments carried on at the testing plant of the Buffalo Forge Co. The first set of experiments was conducted to determine the effect obtained by changing the radius of curvature of the elbow. As might naturally be expected, the loss through a sharp-turn elbow is the greatest, and the loss decreases as the radius of curvature increases. With the radius equal to 1½ diameters of the duct, fairly good results were obtained without making the elbow unduly long; practically nothing was gained by making the radius greater than twice the diameter. The second series of tests was made to determine the effect of reducing the area of an elbow at the throat to the theoretical or equivalent area, the object being to prevent eddy currents along the inner side of the pipe. One source of friction would thus be eliminated and the long-cone effect would aid in changing the accelerated velocity pressure at the turn of the elbow back to static pressure with a minimum loss. With this construction the loss for a standard elbow was reduced as much as 60 per cent.

A paper by Frederick Bass, "Experiment in School Room Ventilation with Reduced Air Supply through Individual Ducts," was the last of the session. In the experiment the air was supplied to each individual desk. During the entire three weeks the test lasted the air was exhausted from the room and used over and over again. Between the exhaust and delivery fans there was an ozone generator and in the equipment there was also provided an air washer and humidifier. The effect on the pupils was closely studied. Each pupil was supplied with about 8½ cu.ft. per min. of renewed air and they were compared mentally and physically with a similar group of pupils receiving 30 cu.ft. of air per min. in an adjoining school. The test results were slightly in favor of the group receiving the smaller amount of air, but it was the opinion of those discussing the paper that although the test was interesting, observations should extend over a much longer period of time before any definite conclusions could be drawn.

TECHNICAL EXCURSIONS

On Friday morning the society as a body visited the Institute of Thermal Research conducted by the American Radiator Co. F. B. Howell, assistant general manager of sales, welcomed the visitors and turned them over to J. H. Davis, L. C. Soule and M. A. May to be conducted through the laboratory. Tests in charge of Dr. C. B. Thompson and L. B. Cherry were being conducted on two house-heating steam boilers, two hot-water houses-heating boilers and on pipe coils under fan blast conditions. Another test was under way to determine the proper size of pipe for one-pipe low-pressure steam systems. A number of radiators containing different amounts of surface were connected by pipes of different diameter to a common supply main carrying 2 lb. pressure. Glass nipples about 12 in. long were inserted in the risers so that any disturbance between the ascending steam and the returning condensation could be noted. The laboratory was well equipped with instruments and facilities for testing all kinds of heating apparatus and the opportunity to look over and inspect this equipment was greatly appreciated by the society.

An interesting afternoon was spent at the plant of the Buffalo Forge Co., J. I. Lyle and W. H. Carrier were there to tell the visitors all about fans, humidifiers, air washers, the various instruments needed in the work and incidentally the method of manufacturing the Spiro turbine.

LAST SESSION

On Saturday there was only a morning session. Professor Allen, past president of the society, who has been in Turkey for the past two years, was present and made a few brief remarks on the progress made by the society. In the early days everything was done by rule of thumb. In the last ten years, however, scientific developments have called for technical men who in the course of time will develop heating and ventilating from a mechanics' art to a science. The first thing necesary is scientific data. The society is beginning to get these data, but there are many things to be learned and a wide field for investigation. For ventilating a room there are no standard methods. It is up to the physiologist to tell first what constitutes proper venilation, and the engineer will then readily fulfill the requirements. Even the amount of radiation required is not definitely settled. It is apparent that there is a tremendous amount of work to be done before heating and ventilation may truthfully be classed as a science.

In England the heating is done mostly by hot water. Practically nothing is known about the single-pipe system of steam heating. The engineers there freely admit that American has gone farther in the development and operation of heating apparatus. In Germany they have developed the theory, but when it comes to practice and installation they too look to America for the latest and best. In doing the work necessary to develop heating and ventilation to a science, Professor Allen looks to a great future for the society.

I. Morgan Bushnell, president of the National District Heating Association, was present and addressed the meeting briefly, referring particularly to the joint committee appointed by the two societies. With the practical knowledge of one and the theoretical knowledge of the other, it was hoped that the combination would produce good results in the solution of the many problems that are now coming to the front. The recent improvements in meters will help greatly in getting valuable data for the solution of some of these problems.

A valuable paper containing a chart for determining the size of pipe for gravity hot-water heating systems was then read by M. S. Cooley. There have been charts for air and steam piping in which all the variables involved could be easily traced, but not for hot water. The method by which the present chart was devised was discussed by the author and its use was illustrated by the solution of several examples.

In a paper on "Heat Transmission with Pipe Coils and Cast-iron Heaters under Fan Blast Conditions," L. C. Soule described a series of 14 tests made at the Institute of Thermal Research of the American Radiator Co. on pipe spaced on 2¾-in. centers. Tests were made on five sections, three sections and one section deep, each section containing four staggered rows of pipe containing 193.1 sq.ft. of surface. The outfit used was placed under test the day before while the society was in the laboratory. The results of the pipe coil tests are all tabulated and the equivalent values in vento heaters given. In comparing pipe coils on 2¾-in. centers with vento heaters it was shown that for the same friction 35 per cent. greater velocity is obtained through the vento than through the pipe coils. This result was obtained when using one stack deep of regular vento on 5⅜-in. centers of loops against each section deep of four-row pipe coils.

Chicago is the first city to control the ventilation of buildings, which is done by the ventilation inspection department of the bureau of health. Dr. E. Vernon Hill, who

organized and is conducting this department, gave an interesting talk on his work. Dr. Hill briefly summarized the ordinances as applied to theater ventilation, told how the inspections were conducted and showed by means of lantern slides the card records and some typical installations.

D. D. Kimball gave a list of the members of the ventilation commission of New York City, which was recently appointed by Governor Sulzer and provided with a fund of $59,000. He outlined the scope of the work and described the facilities they would have for conducting experimental and research work.

Routine business and some resolutions of thanks to companies which had extended courtesies to the society ended the session and the convention. It may be of interest to note that the register showed an attendance of 133, of which 75 were members.

GENERAL EXCURSIONS

For entertainment the local committee had provided a good program. Diversions for the ladies in the form of sight-seeing automobile trips in and around Buffalo, including visits to Niagara Falls and to the Roycroft industries established by Elbert Hubbard, were arranged. On Thursday evening the entire delegation had a boat trip to Crystal Beach and on Friday night another boat trip to Grand Island where dinner was served at the Bedell House. A feature was an interesting talk by Professor Allen on conditions in Turkey and the characteristics of its inhabitants. On Saturday afternoon some of the conventioners visited Niagara Falls and the plant of the American District Steam Co., which is in that vicinity.

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Massachusetts N. A. S. E. Convention

The annual Massachusetts State Convention of the National Association of Stationary Engineers took place at Lawrence, Mass., July 10, 11 and 12. The Franklin Hotel was the headquarters. The mechanical exhibit was held in the Armory, which was tastefully decorated and arranged for the purpose. Fifty booths were occupied. The exhibition was formally opened at noon on Thursday. Addresses were made by Albert C. Ashton, president of the New England Association of Commercial Engineers; Claude D. Allen, chairman of the exhibit committee, and State Vice-President Asa M. Day.

On Friday morning at ten o'clock the preliminary exercises of the convention were held in the city hall, R. N. Howard, chairman of the arrangement committee presiding. There were about 70 delegates in attendance. Mayor M. F. Scanlon welcomed the convention to the city of Lawrence, and National Vice-President James R. Coe responded for the engineers. Addresses were also made by F. N. Chandler, Theodore N. Kelsey, Edward H. Kearney, Charles H. Sumner and the Hon. William McComb. State President John T. Maloney then took charge of the meeting, and after the necessary committees were appointed the session adjourned.

The program of entertainment included auto trips about the city, a lecture on the steam turbine by Elmer Smith, of the General Electric Co., and a smoker at the City Hall, where fully 400 engineers and guests gathered. On Saturday afternoon special cars were boarded for a trip to Canopie Lake, where an appetizing banquet was served. The game of baseball between the engineers and supplymen was won by the latter with a score of 12 to 6. The plants of the American Woolen Co. mills were at the disposal of the convention through the invitation of George H. Diman, consulting engineer.

At the final session of the delegates the following state officers were elected: Asa M. Day, president; Thomas Ray, vice-president; John F. Quinn, secretary; Walter Damon, treasurer; Andrew Waite, conductor; Merrick M. Childs, doorkeeper.

The selection of the officers of the New England Association of Commercial Engineers resulted as follows: Bradford L. Ames, president; W. Carleton Barnes, vice-president; Herbert E. Stone, treasurer; Albert C. Ashton, John A. Stitt and F. P. Upson, directors.

During the convention, Albert C. Ashton was presented a handsome traveling bag and toilet set, and Claude D. Allen received a gold watch fob.

The next meeting will take place at Providence, R. I., in July, 1914.

The following is a list of the exhibitors:

Lawrence Machine Co.; W. R. Winn; McLeod-Henry Co.; Manning, Maxwell & Moore; B. L. Ames; George W. Knowlton Rubber Co.; L. A. Couch; Charles A. G. Winther; Hamblett Machine Co.; Wickes Boiler Co.; "Practical Engineer"; Cancos Manufacturing Co.; Philadelphia Grease Manufacturing Co.; Griscom-Russell Co.; Eagle Oil & Supply Co.; J. H. Blanchard; H. W. Johns-Manville Co.; Curry Brothers Oil Co.; Lunkenheimer Co.; Garlock Packing Co.; Bride-Grimes Co.; Economy Lubricating Co.; Autoforce Ventilating System; American Tool & Machine Co.; Calvin H. Currier; Greene, Tweed & Co.; Jenkins Bros.; Ashton Valve Co.; Mason Regulator Co.; International Engineering Works; U. S. Graphite Co.; A. M. Byers Co.; Pennsylvania Flexible Metallic Tubing Co.; Hart Packing Co.; Lumsden & Van Stone Co.; Durabla Manufacturing Co.; Under-Feed Stoker Co. of America; Adam Cook's Sons; Chicago Pneumatic Tool Co.; Robinson Hardware Co.; A. W. Chesterton Co.; Autogenous Welding Equipment Co.; American Steam Gauge & Valve Manufacturing Co.; American Steam Packing Co.; Diamond Power Specialty Co.; F. W. Goldwait; Keystone Lubricating Co.; Evans Mill Supply Co.; Lagonda Manufacturing Co.; Patterson Lubricating Co.; Treat Hardware Co.; Watts Regulator Co.; The Philip-Carey Co.; Enterprise Rubber Co.; "Power"; "Southern Engineer"; "National Engineer"; Archibald Wheel Co.; Callison Brass Foundry Co.; Dearborn Chemical Co.; Monarch Valve & Manufacturing Co.; Crandall Packing Co.; Gardner Gate Co.; A. N. Little Co.; W. J. Hyland.

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Wisconsin N. A. S. E. Convention

The Wisconsin state association of the National Association of Stationary Engineers held its thirteenth annual convention at Milwaukee, on July 10, 11 and 12. Fred C. Ruck, chairman of the local committee, called the first session to order at 10:30 a.m., on Thursday and introduced the Honorable G. A. Bading, mayor of the city, who welcomed the delegates and guests to Milwaukee. In the course of his address, the mayor said that it seemed fitting for the engineers to meet in Milwaukee because of the city's extensive industrial activities and the important part played by the engineer in practically all forms of modern industry.

After mentioning the fact that Milwaukee has an effective engineers' license law, he said he believed that a state law would be preferable as, with such, all engineers would be on an equal basis, while at present engineers coming to Milwaukee from other points must submit to examination before they may operate in the city.

State-President Henry J. Mistele, the next speaker to be introduced, pointed out that it was often difficult for the engineer to get away from his duties long enough to attend a convention and this fact is one of the main reasons why the conventions are not far larger in size. The necessity for the engineer's close application to his duties often works a hardship, as many a man, through thus being confined, has failed to develop all the latent possibilities he possesses, simply because he could not obtain the finish which is to be secured only by contact with other men.

President Mistele also commented on the educational value of state conventions due to the papers and lectures invariably included in the program and the exhibit by the machinery and supply concerns.

National President John F. McGrath said that engineers, like the men in every other line of endeavor, must "make good" to win proper recognition and financial encouragement. He said there were two types of men he did not like. First, the man who takes you down into his engine room, and gives you a long harangue on what a wonderful plant and operating engineer are before you, winding up with the intimation that his only regret is that some day he will die and the boss will have to go out of business because he will not be able to secure anyone to fill the engineer's place. Second, the man who is so lacking in self-confidence that he is afraid that if he loses his position it will kill him because he will never be able to get another one. Dismissal is often a blessing in disguise to such men, for it is the only way in which they would ever be spurred to greater effort.

Charles Cullen, secretary of the Central States Exhibitors' Association, spoke briefly on the exhibits.

Fred W. Raven, national secretary of the N. A. S. E., discussed Milwaukee as the scene of the 1914 national convention. He also touched upon the subject of biennial national conventions instead of annual events, pointing out that the money saved in mileage disbursements to the delegates could well be used for other purposes such as for the educational work, accelerating the enactment of license laws, etc. He stated that the association, just like any business house, must be uptodate and employ modern policies and methods if it is to endure and prosper.

John W. Lane, editor of the "National Engineer," said that an important feature of conventions and meetings is that they bring the individual into contact with his fellow men and help to round him off. Another phase is the opportunity pre-

sented of practicing the expression of his thoughts and ideas while on his feet and in the presence of others. No matter how able a man may be as far as his vocation is concerned, he must be capable of displaying his capabilities to good advantage on occasion if he is to reap fullest benefit from them.

At the Friday afternoon session, J. L. White gave a short paper on "The Operating Engineer as a Political Factor." By way of introduction he explained that he did not favor extreme political activity on the part of the engineer except when laws were involved that related to his occupation. It was asserted that legislation is badly needed to provide for the proper regulation of consulting engineers. Mr. White stated that the public was too often duped by men setting themselves up as consulting engineers when they had little or no suitable qualifications. Where ignorant or inexperienced consulting engineers are employed to lay out or revise plans for a power plant, the operating engineer is often handicapped and placed under a hardship by bungling work. Hence, all operating engineers could profitably concern themselves with the bringing about of suitable corrective legislation.

In the evening, John W. Lane, assisted by Alfred Johnson, gave a lecture on "The Panama Canal," in Miller's Hall. The lecture was illustrated with more than 70 stereopticon views of the various parts and features of the canal, as well as some of the towns and settlements connected with it. Dancing followed the lecture and light refreshments were served.

At the Saturday morning session Prof. Robt. Bauer gave an interesting lecture on "Coal Analysis and the Purchase of Coal on a B.t.u. Basis." The lecturer demonstrated the process of analysis with actual apparatus. Wm. Classman lectured on "The Utilization of Exhaust Steam."

Among the important matters of association business taken up was the question as to whether biennial national conventions should or should not supplant the annual event which is at present customary and a resolution was passed favoring the change.

The officers elected for the present year are A. Gunderson, Sheboygan, president; Fred. Ruck, Milwaukee, vice-president; Robt. Fenn, Sheboygan, reëlected secretary; J. J. Novotny, Madison, treasurer; J. A. Wickert, Milwaukee, state deputy.

The exhibit was held in a large room in the hotel and was, as usual, under the direction and management of the Central States Exhibitors' Association. More than 45 displays were made, some of which were of unusual interest.

While plenty of business and educational work was provided for, the entertainment features were not neglected. The ladies enjoyed an automobile ride on Friday afternoon and a shopping and sight-seeing tour on Saturday morning. On Saturday evening the Exhibitors' Association gave its annual "smoker." Quotation marks are used because the ladies were present and the function assumed so few of the usual characteristics of a smoker that the term is all but a misnomer. Interspersed with addresses by the prominent members of both organizations were elocutional and vocal selections by the entertaining Mr. Fred Hickey, of the Dearborn Chemical Co., his vocal efforts being assisted at times by all present joining in the chorus.

On Sunday the engineers held their annual picnic at Lincoln Grove, Layton Park, where the ball teams representing the engineers and the supply men met in their annual "clash," as the sporting editor would express it, for the punch-bowl trophy donated two years ago by H. S. Bowers. The score was 11 to 9 in favor of the engineers, who, thereby, won the trophy from the supply men who were victorious last year. Other features of the picnic were races and games for boys, girls, women, fat men, engineers, supply men, etc., and a tug-o'-war between the engineers and supply men, which was won easily by the former.

The place for the next convention will be selected later by committee.

OBITUARY

Anthony N. Brady died in London, England, on July 22. Mr. Brady was a selfmade man, and from a very humble beginning rose to be a power in the financial world and the head of many large corporations. Among them were the New York Edison Co., the Edison Electric Illuminating Co., Brooklyn, and the Kings County Electric Light & Power Co., and many more. He was chairman of the board of directors of the Brooklyn Rapid Transit Co., trustee of the Consolidated Gas Co., of New York, director of the Hudson & Manhattan R.R. Co., and director of the United States Rubber Co. Mr. Brady was born in France of Irish parentage, 70 years ago. He leaves a widow, two sons and three daughters.

BUSINESS ITEMS

The American Engineering Co., Philadelphia, has just issued three very interesting booklets entitled, "Morning," "Noon" and "Night," in connection with the Taylor stoker publicity. Copies are sent free on request to anyone interested in stokers.

A very interesting booklet has recently been issued by Greene, Tweed & Co., 109 Duane St., New York, which shows some of the buildings in and around New York in which Rochester automatic lubricators are installed. It's worth sending for.

DELEGATES AND VISITORS AT THE WISCONSIN STATE N. A. S. E. CONVENTION, MILWAUKEE, JULY 11, 12 AND 13, 1913

The Griscom-Russell Co., 90 West St., New York, has established its own New England office in Boston, 141 Milk St., to take care of the growing demand for Griscom-Russell specialties in New England. Mr. Wm. A. Evans has been appointed manager of the new office.

Owing to the increase of their business in New England, Schutte & Koerting Co., 12th and Tompson Sts., Philadelphia, have moved their Boston office from 98 High St. to larger quarters at 132 High St. The Denver office has also been moved from 1710 Glenarm St., to the First National Bank Bldg.

The American Pulley Co., Philadelphia, has moved its New York branch from 203 Lafayette St. into more commodious quarters at the corner of Grand and Green Sts. At the new address is carried a complete stock of "American" belt pulleys. Orders will be received for reels, spools, beams, sash pulleys, and pressed-steel shapes.

The Western Kieley Steam Specialty Co., 116-122 West Illinois St., Chicago, has published a new 96-page catalog illustrating and describing its various steam, air and water regulating appliances. In addition to the Western-Kieley goods, the company has added the Boylston line, and this is fully described in the new catalog, a copy of which is sent free on application.

"A Collapsible Oil Filter" is the title of a new booklet published by S. F. Bowser & Co., Fort Wayne, Ind., illustrating and describing the Bowser oil filtering and circulating